DOE /NASA CONTRACTOR REPORT

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STATUS REPORT ON PRELIMINARY DESIGN ACTIVITIES FOR SOLAR HEATING AND COOLING SYSTEMS

Prepared by

AiResearch Manufacturing Company
2525 West 190th Street
Torrance, California 90509

Under Contract NAS8-32091 with

National Aeronautics and Space Administration
George C. Marshall Space Flight Center, Alabama 35812

For the U. S. Department of Energy

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Solar Energy

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16. ABSTRACT

Information presented provides status and progress made by AiResearch Manufacturing Company on the development of solar heating and cooling systems. The major emphasis is placed on program organization, system size definition, site identification, system approaches, heat pump and equipment design, collector procurement, and other preliminary design acitivities as part of the contract requirements.

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1. INTRODUCTION AND SUMMARY

1.1 INTRODUCTION

This document summarizes the preliminary design activities conducted by AiResearch Manufacturing Company of California under Contract NAS8-32091. It was prepared in accordance with the requirements of Appendix A, Data Requirement 500-7 of the Statement of Work.

The activities reported here were conducted in the period from July 12 (effective date of the contract) to September 1, 1976. During this period major emphasis was placed on the following:

(a) Program organization

(b) Heating and heating/cooling system size definition

(c) Site identification

(d) Refinement of system approaches

(e) Refinement of the heat pump design

(f) Heat pump equipment preliminary design

(g) Preparation of an RFP for collector procurement

(h) Generation of program data as specified in Appendix A of the Statement of Work

1.2 PROGRAM ACTIVITIES

A kickoff meeting was held at AiResearch on July 27, 1976. Present at the meeting were NASA program management personnel and AiResearch solar program personnel. A number of action items were agreed upon; the most significant of these in terms of impact on the PDR schedule and scope are listed below.

- AiResearch/Dunham-Bush has agreed to conduct a market analysis to define optimum sizes for the marketable systems. As a result of these analyses, the following heat pump sizes were identified (the capacities given correspond to the standard ARI rating conditions for heating and cooling).

  (a) Single-family residence

  Cooling: 3 tons
  Heating: 60,000 Btu/hr

  (b) Multifamily residence

  Cooling: 25 tons
  Heating: 600,000 Btu/hr

  (c) Commercial application

  Cooling: 10 tons
  Heating: 200,000 Btu/hr

  The system maximum capacity can be enhanced through the use of auxiliary heaters. The ratio of heating to cooling capacities far exceeds that achieved with present-day heat pumps. The rationale for size selection and the results of the marketing survey are presented in Appendix A. Heat pump preliminary design is proceeding using the sizes listed above.

- AiResearch is to conduct an investigation to identify sites suitable for long-term evaluation testing of the 12 systems. Detailed definition of the systems and performance characterization will proceed following site selection and analysis. Actual site data are essential for system optimization and collector and storage tank sizing in terms of energy conservation and present value cost. The analytical tools are available to perform this task; they will be used at a later date. For the purpose of completeness, the system performance

data given in the AiResearch proposal are presented here for the single-family and multifamily heating systems. These data were generated using the residence models and the Madison weather tapes supplied by NASA.

A coordination meeting was held at NASA for the purpose of establishing common guidelines for system analysis for all solar system design and development contractors. The content of the NASA data bank was also explained at this meeting. Most of these data, including weather data for the selected sites and cost analysis parameters, will be necessary for system definition.

1.3 PROGRAM DOCUMENTATION

The following documents were prepared in accordance with the requirements of Appendix A of the Statement of Work. These documents are included as part of this preliminary design data package by reference.

(a) Development Plan--DR 500-1--AiResearch Report No. 76-13047

(b) Verification Plan--DR 500-2--AiResearch Report No. 76-12996

(c) Quality Assurance Plan--DR 500-3--AiResearch Report No. 76-13043

(d) Special Handling, Installation and Maintenance Tool List--DR 500-15--AiResearch Letter CAJWY:6601:0825 to K. Sowell, NASA, dated August 25, 1976

(e) Hazard Analysis--DR 500-18--AiResearch Report No. 76-13048

(f) Logistic Plan--DR 500-22--AiResearch Report No. 76-13051

(g) Safety and Health Plan--DR 500-24--AiResearch Report No. 76-13046

(h) New Technology Reporting Plan--DR 500-25--AiResearch Report No. 76-12776

(i) WBS and Dictionary--DR 500-26--AiResearch Report No. 76-13156

(j) Instrumentation List--AiResearch Reporet No. 76-13139

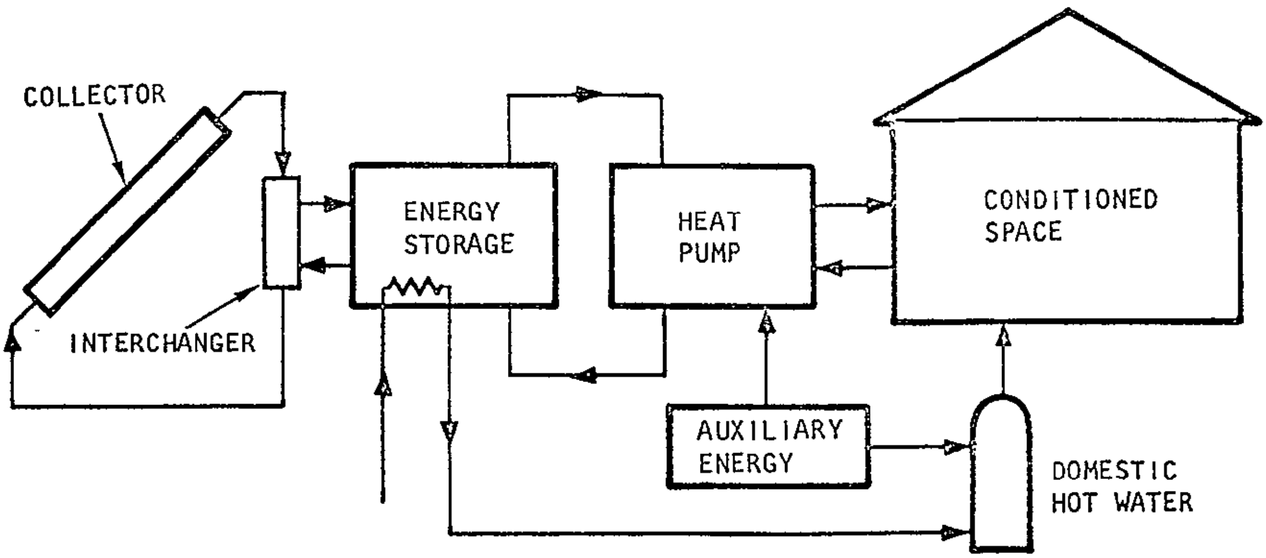
1.4 OVERALL SYSTEM DEFINITION

Figure 1-1 depicts the overall arrangement of the heating and heating/cooling systems. Schematically, both types of systems are the same. This similarity also extends to the design of the heat pump. Most of the equipment of this subsystem (heat exchangers, compressor, compressor motor, and controls) was designed to be optimum for both the heating and cooling functions. This involves a number of design trade studies to ensure high performance in both modes over a wide range of operating conditions. These studies were performed in the definition of the heat pump subsystem; some of the data generated for the cooling mode of operation are included in this report, since they are basic to the design of the heat pump and its components.

1.5 ORGANIZATION

The data presented in this report are arranged in the following fashion.

- Heating system description and performance
- Collector subsystem discussions
- Energy storage design approach
- Heat pump preliminary design
- Auxiliary energy subsystem specifications
- Domestic hot water subsystem discussions
- System control preliminary design
- Heat pump equipment design, including
    - (a) Turbomachine
    - (b) Motor
    - (c) Motor control
    - (d) Heat exchangers
- System/subsystem trade studies



SUMMARY OF APPROACHES

SUBSYSTEM	SELECTED APPROACH
COLLECTOR	FLAT PLATE COLLECTOR
ENERGY STORAGE	WATER TANK STRATIFIED AND OPTIMALLY INSULATED
SPACE HEATING	VARIABLE-SPEED MOTOR-DRIVEN CENTRIFUGAL COMPRESSOR HEAT PUMP
SPACE HEATING/COOLING	SPACE HEATING--AS ABOVE; SPACE COOLING--RANKINE-CYCLE SOLAR-POWERED TURBOCOMPRESSOR AIR CONDITIONER
AUXILIARY ENERGY	SPACE HEATING AND DOMESTIC HOT WATER--NATURAL GAS SPACE COOLING--ELECTRIC MOTOR AUGMENTATION OF THE SOLAR-POWERED TURBINE
HOT WATER	HEAT EXCHANGE WITH ENERGY STORAGE TANK
ENERGY TRANSPORT	PRIMARILY OFF-THE-SHELF EQUIPMENT
CONTROLS	COMMERCIAL TYPE EQUIPMENT
SITE COLLECTION DATA	INSTRUMENTATION SELECTED FOR COMPLETE PERFORMANCE EVALUATION
ELECTRICAL SUBSYSTEM	OFF-THE-SHELF COMMERCIAL HARDWARE

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Figure 1-1. System Approach

2. HEATING SYSTEM APPROACHES

2.1 GENERAL

This section describes the three proposed heating systems. The descriptions are generally given in terms of the various subsystems; however, details on controls, interfacing equipment (part of energy transport subsystem), and heat transport to the conditioned space (part of heat pump subsystem) are presented for intelligibility. More details on the subsystems are presented in later sections of this document.

Complete system definition, including collector and thermal energy storage tank sizing, and cost/performance optimization must be based on site parameters to determine the loads, insolation, and energy cost. Since the sites have not been selected yet, the system analyses conducted used the residence models and the weather data supplied by NASA for purposes of proposal evaluation. These system-level trades were aimed at optimization of the system/subsystem arrangement and design point parameters; they are reported in Section 12 of this document.

No other effort was expended in system definition; therefore, the performance data presented were taken from the AiResearch proposal to NASA (AiResearch Report 75-12313-1) and are shown here for reference purposes only.

Performance data are presented for the single-family and multifamily residence systems only; no system-level performance data are available at this time for the commercial size heating system.

2.2 SYSTEM OVERVIEW

The heating system designs for the three applications under consideration use the same subsystem approaches and are schematically almost identical. To simplify the description of these systems and to obviate unnecessary repetition, an overview of the major subsystem approaches is presented first. It is important to note here that no significant changes have been incorporated into the system design since the proposal.

2.2.1 Collector Subsystem

Preliminary subsystem-level studies conducted prior to contract award indicate that flat-plate, double-glazed collector panels without selective coating were optimum for use with the variable-speed heat pump. In August 1976, an RFP was issued by AiResearch, and proposals have been requested from 31 collector manufacturers. Data furnished by the bidders will be evaluated in terms of design features, durability, potential corrosion problems, behavior under stagnation conditions, thermal performance, impact on system/subsystem configuration, availability, and cost. All types of collectors will be evaluated using these criteria.

A schematic of the collector subsystem as currently conceived is presented in Figure 2-1. This collector loop was developed assuming that the collector can repeatedly be subjected to stagnation temperature without damage. Water is the collector fluid.

The collector loop is closed; it interfaces with the remainder of the system through a steel interchanger. In this manner, materials in the collector loop can be controlled better; and corrosion can be avoided through the use of suitable commercial inhibitors or deionizers. The presence of the interchanger will result in slightly higher collector temperatures and larger collector areas; however, these penalties may well be offset by the

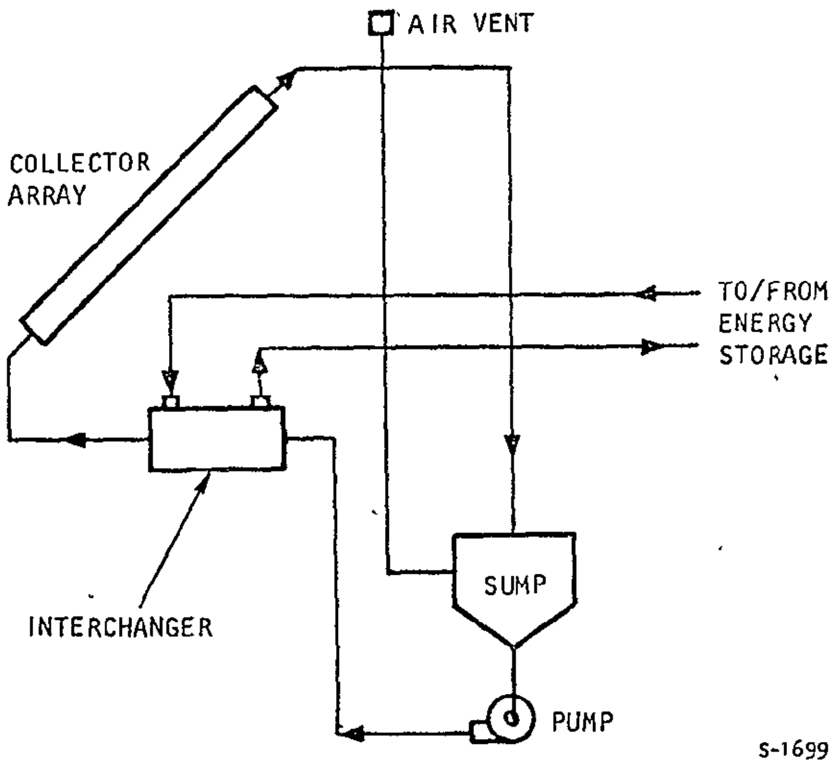

Figure 2-1. Collector Loop Arrangement

corrosion protection afforded by its use. Detailed evaluation of the interchanger will be conducted with the candidate collector evaluation (see Section 12).

Freezing of the collector water is prevented by draining the collector panels, manifolds, and lines into a sump located within the confines of the conditioned space where there is no danger of freezing. Draining will occur automatically upon deactivation of the collector loop pump. Air contained in the sump when the collector is full of water will be displaced into the collector panels during the draining operation. An air vent is provided to balance the loop pressure. Usually, only a very small quantity of fresh air is vented out of and into the loop during normal operation, so the oxygen intake of the loop is minimized.

2.2.2 Heat Pump Subsystem

The heating subsystem features a motor-driven vapor compression heat pump. For cooling, the same vapor compression loop is used; however, in this case a Rankine power loop is incorporated into the package. The Rankine power loop turbine then drives the compressor. Both heating and heating/cooling subsystems are described below.

2.2.2.1 Heating Subsystem

A motor-driven vapor-cycle heat pump provides the means of raising the temperature level of low-grade thermal energy stored in the water tank to a level suitable for space heating. Thus, solar energy can be collected at a relatively low temperature with a relatively inexpensive collector operating at reasonable efficiencies. A simplified schematic of the heat pump is depicted in Figure 2-2, with an illustration of the thermodynamic processes occurring within the working fluid loop. Solar energy transferred to the working fluid at the evaporator is rejected to the residence at the higher temperature level of the condenser. The compressor is used to maintain this temperature differential. R-11 was selected as the working fluid because of its superior thermodynamic properties, especially for the smaller size subsystem (see Section 12).

The proposed heat pump utilizes a centrifugal compressor driven by a motor controlled so that compressor speed and work are maintained at a minimum value consistent with the capacity requirements and the temperature level of the heat source. In this manner, motor input power is always minimized and heat pump COP is maximized. In the small-size design, for example, the heat pump COP (heat delivered/total electrical energy input) at design point (60,000 Btu/hr and 60°F water temperature) is estimated at 4.0; at a water temperature of 90°F and 2/3 design load, the same heat pump will be operated at a COP of 8.4.

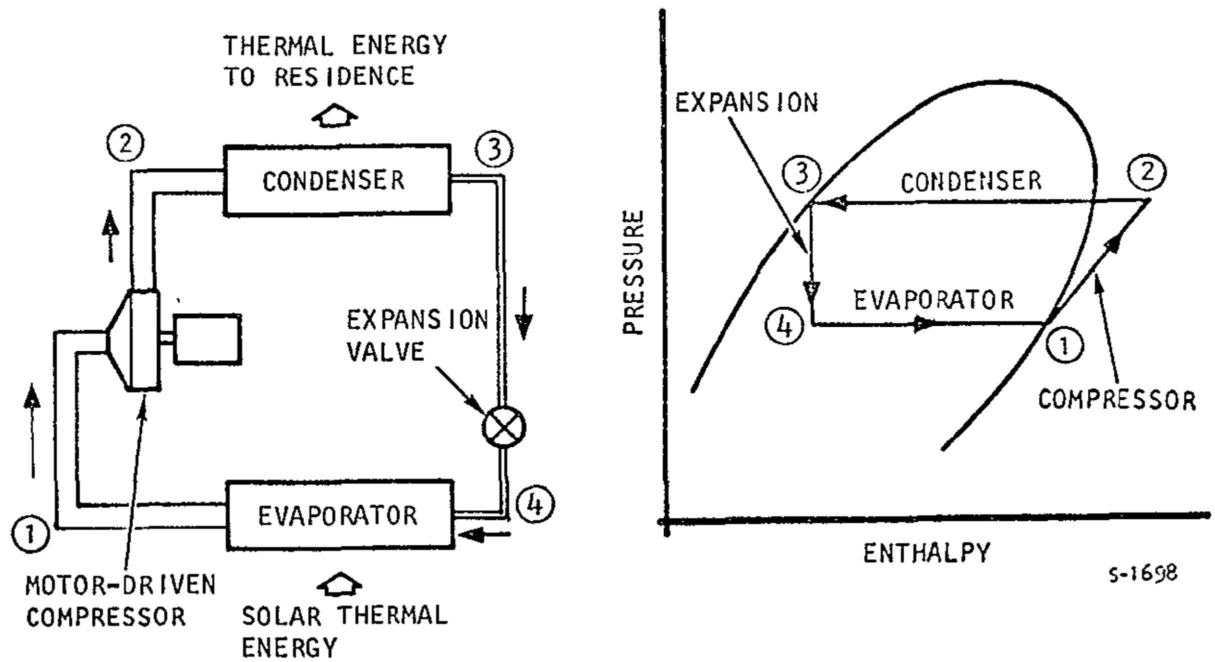

Figure 2-2. Vapor-Cycle Heating Subsystem

Compressor speeds at these operating points are 82,600 and 50,800 rpm, respectively.

The electronic circuitry involved in providing speed control capabilities over a wide range utilizes standard circuitry and detail parts. The cost of the circuitry is partially offset by the savings effected through the use of small high-speed motors directly coupled to the turbomachines.

The proposed motor-driven compressor features foil bearings and does not require any lubricant other than the process fluid. This in itself eliminates potential development problems involved with the use of ball bearings. Further, heat exchanger performance is enhanced by elimination of the lubricant.

2.2.2.2 Cooling Subsystem

The cooling subsystem (Figure 2-3) utilizes solar thermal energy to power a turbine through the Rankine cycle. The power developed by the turbine is

used to drive the centrifugal compressor of a vapor compression refrigeration system. The same working fluid, R-11, is used in both the power and refrigeration loops so that a common condenser can be used. Sealing problems at the turbocompressor are thus obviated. An electric motor integral with the turbocompressor is used for system operation when the solar energy source is inadequate to power the compressor.

Using switchover valves and isolating the Rankine power loop, the cooling subsystem can easily be converted to the heating subsystem shown in Figure 2-2.

The compressor, motor, condenser, and evaporator of the heating/cooling systems are identical to those of their heating-only system counterparts. In addition, the motor control electronics and the system control modules are the same. For this reason, the design of both heating and heating/cooling systems must be presented in parallel to assure optimum performance in both modes of operation.

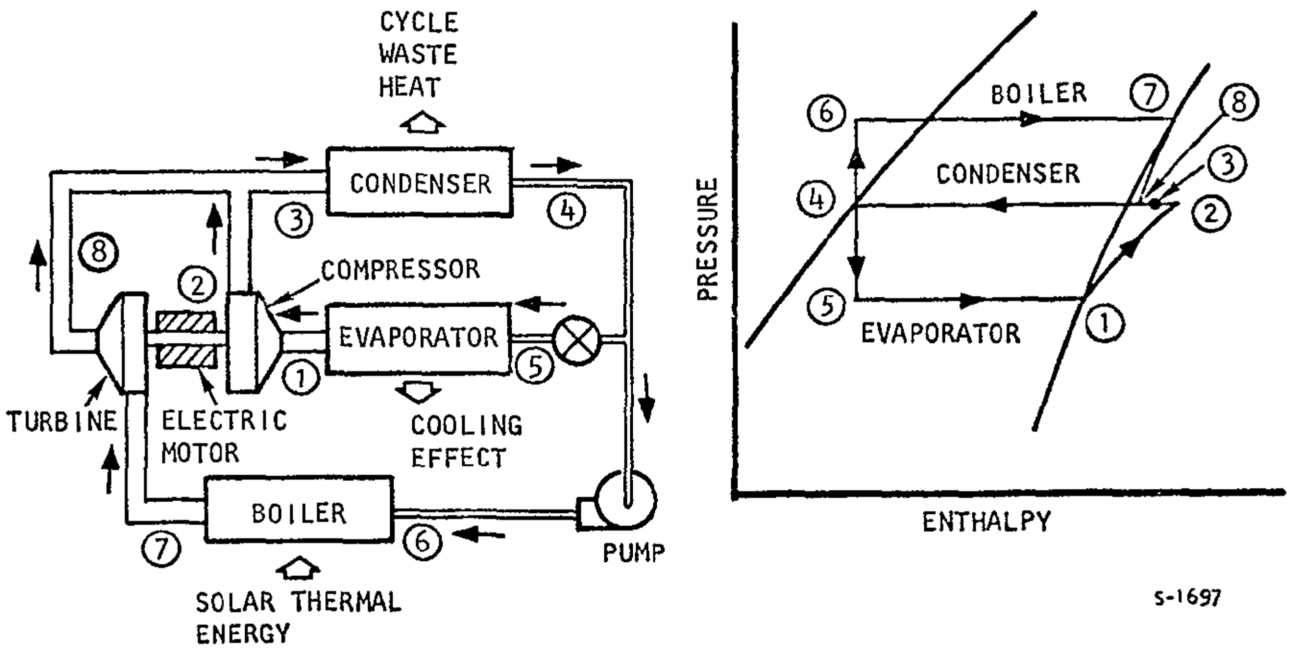

Figure 2-3. Cooling Subsystem Arrangement

2.2.3 Thermal Energy Storage

Water is used as the thermal energy storage medium for reasons of cost, availability, and development status.

2.2.4 Auxiliary Energy

It is assumed that natural gas will be available at the test sites. All heating systems use off-the-shelf equipment for auxiliary heating.

2.3 SYSTEM DESCRIPTIONS

2.3.1 Single-Family Residence Heating System

The schematic in Figure 2-4 shows the arrangement of this system. Only the sensors necessary for system operation are shown; for simplicity, instrumentation necessary for performance evaluation is not included.

2.3.1.1 Collector

The overall features of this subsystem have been described previously in para. 2.2.1.

In actual installation, the sump should be located near the base of the collector consistent with proper draining of the collector. In this manner, the heat developed by the pump will be minimized.

The collector and storage tank pumps are started at a signal from the collector control module. This module compares the temperature of the collector at a strategic location on the array to that of the water in the thermal energy storage tank. When the collector temperature exceeds that of the water, a relay is activated and the two pumps are started. The collector will fill with water, and circulation will be maintained through the collector.

During summer operation, storage tank temperature will be limited to 180°F to prevent boiling in the tank and in the collector. When the tank temperature reaches that value, the pump is deactivated and the collector is drained.

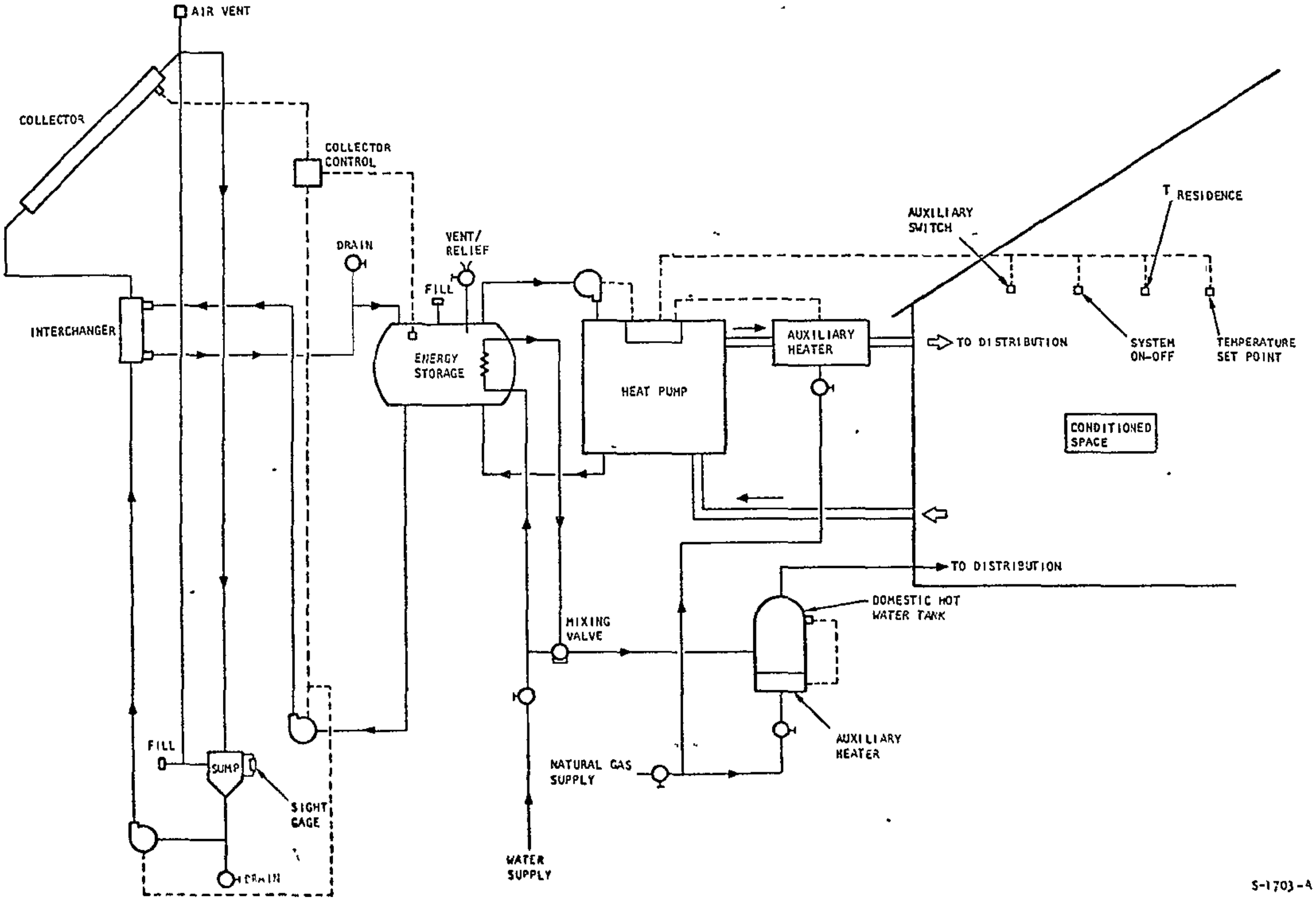

Figure 2-4. Single-Family Residence Heating System

2.3.1.2 Water Storage Tank

The tank water content is estimated at 1200 gal. This value was determined from tradeoff studies presented in the AiResearch proposal. The entire tank is fabricated of steel and will be insulated with 14 in. of fiberglass insulation (k = 0.24 Btu/hr-sq ft-°F/in.) or equivalent.

The actual location of the tank with respect to the collector will be a major consideration once the demonstration sites are selected. Careful attention will be paid in order to prevent deterioration of the thermal properties of the tank insulation through water absorption. This also applies to all the hot water lines of the collector, tank, and heat pump circuits.

Valves are provided on the tank to (1) permit draining of the tank using the pump (depending on installation constraints), (2) relieve excess pressure that could build up within the tank, and (3) fill and vent the tank as required.

2.3.1.3 Domestic Hot Water

The domestic hot water supply is first circulated through a tube coiled around the hot water tank, where it is heated to within 5°F of the stored water. This water then flows through a mixing valve that maintains the temperature of the hot water to the domestic hot water storage tank at a maximum of 140°F.

The use of the thermal energy storage tank for purposes of domestic hot water heating requires operation of the collector subsystem in the summer. However, the low thermal requirements of this subsystem relative to the size of the collector will ensure high storage tank temperatures in the summer so that no auxiliary energy will be necessary for the domestic hot water supply.

The domestic hot water tank is basically an off-the-shelf, gas-fired, 50-gal tank with a burner capacity of 60,000 Btu/hr. Additional insulation will be required to minimize heat leaks.

2.3.1.4 Heat Pump

The heat pump is a vapor-cycle machine that utilizes the low-temperature heat from the thermal energy storage to vaporize the R-11 working fluid, which is then condensed at the higher temperature level corresponding to the compressor outlet pressure. Air recirculated from the residence is the heat sink for the heat pump condenser and is used to carry the heat pump effect to the residence. The variable speed feature of the heat pump (discussed in more detail in Section 5) permits operation at maximum COP under all conditions.

2.3.1.5 Auxiliary Heater

The auxiliary heater is an off-the-shelf gas-fired furnace currently marketed by Dunham-Bush. This unit will deliver 80,000 Btu/hr. The efficiency of the unit as measured in test is 75 percent excluding losses associated with installation and thermal cycling.

2.3.1.6 Controls

Collector circuit controls have been described above. The controls associated with operation of the heat pump include: (a) a residence temperature sensor located at a suitable location in the residence, (b) a residence temperature selector, (c) an on-off switch to control the entire system, and (d) an auxiliary on-off switch to control operation of the auxiliary heater.

The information provided by these instruments is used by the heat pump control module to vary the speed of the heat pump compressor so that the temperature of the residence matches the desired set point.

Controls associated with the operation of the auxiliary heater and the domestic hot water storage tank are internal to the off-the-shelf packages. They include all interlocks imposed by safety considerations and required by code.

2.3.2 Multifamily Residence Heating System

The multifamily residence heating system is very similar to the 80,000-Btu/hr unit described above. Figure 2-5 shows the schematic arrangement. To obviate repetitive descriptions, the differences between the 800,000-Btu/hr system and the smaller unit are discussed below.

2.3.2.1 Collector

The basic collector panel is the same as for the single-family unit. In this case, the active collector area will be considerably larger for optimum cost and energy savings. The actual arrangement of the collector cannot be definitized at this time; however, it is most likely that the collector panels will be assembled in a number of banks. The schematic of Figure 2-5 illustrates two such banks. The collector loop and control are schematically the same as described previously. Adequate spacing is assumed to preclude shadowing effects.

2.3.2.2 Energy Storage

The energy storage is in a heavily insulated water storage tank. An underground gunite tank may be optimum for this application, depending on the peculiarities of the site. Several steel tanks may be used alternately. In this case, the tanks will be balanced carefully for parallel operation.

Once the actual installation details are available, an evaluation will be made of the benefits that could be derived from scheduling tank thermal energy management to optimize storage temperatures in terms of overall system operation. The controls required for such an operation will be determined.

2.3.2.3 Heat Pump

The 800,000-Btu/hr heat pump is similar to that described previously. In this case, however, an intermediary water loop is used to transport the heat generated at the heat pump condenser to terminal units within the

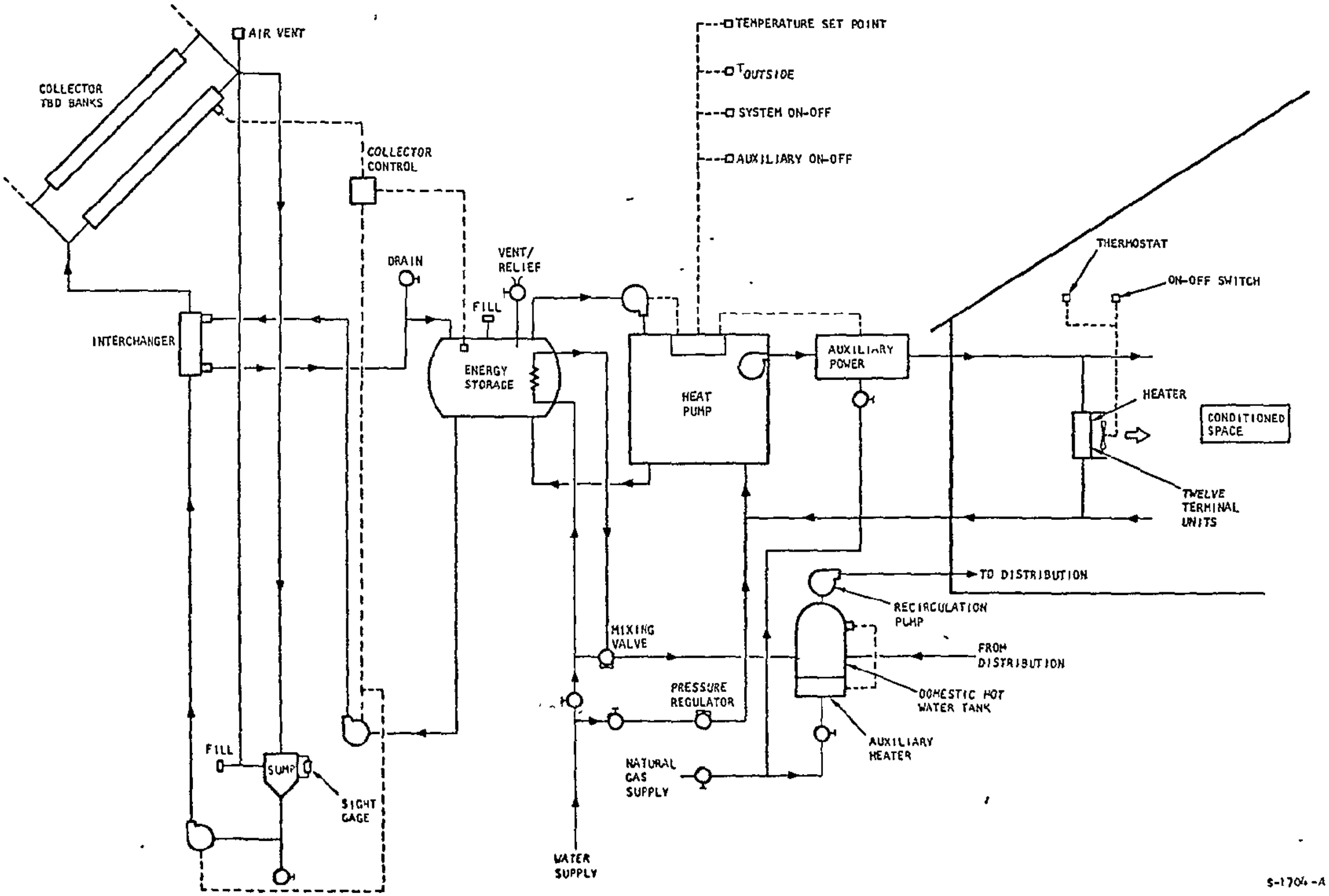

Figure 2-5. Multifamily Residence Heating System

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conditioned space. The actual number of terminal units will be determined following site selection.

An expansion tank with appropriate relief valve, pressure regulator, and fill and drain connections is provided as part of the auxiliary boiler in the water recirculation loop.

2.3.2.4 Auxiliary Heater

The auxiliary heater is a gas-fired boiler currently marketed by Dunham-Bush under the Iron Fireman trade name. This unit has an 800,000-Btu/hr capacity. The efficiency of the auxiliary heater has been demonstrated to be higher than 80 percent excluding installation losses.

2.3.2.5 Domestic Hot Water

The high delivery and recovery rates specified for the multifamily residence require that a separate 650-gal tank be used with a 750,000-Btu/hr gas-fired heater. This equipment is available commercially. The distribution system in this case incorporates a recirculation pump to the many use points of the multifamily residence.

2.3.2.6 Controls

System control is based on essentially the same principle as that used for the 80,000-Btu/hr unit, whereby the heat pump speed is controlled at a minimum value to satisfy the load while operating at maximum COP. There are some minor differences, however, in the details of control. In this case, an on-off switch and a thermostat are provided at each terminal unit and control operation of the ventilation fan of that particular unit. Hot water flow to the terminal units is controlled from the heat pump.

Heat pump control is effected with an on-off switch, an auxiliary switch, a temperature set point, and a sensor measuring outside air temperature. Other

sensors measuring water temperature within the heat pump subsystem also are used for control, as will be described in detail later (see Section 5).

2.3.3 Commercial Application Heating System (250,000 Btu/hr)

The commercial application heating system is conceptually the same as the single-family residence system shown in Figure 2-4.

2.4 SYSTEM PERFORMANCE

System performance for the single-family and multifamily residence systems is presented in Table 2-1. These data were taken from the AiResearch proposal and were derived using Madison weather data and the residence models supplied by NASA. They are included here for reference purposes only. It is believed that these data are representative of the system designs that will be generated following site selection.

TABLE 2-1

HEATING SYSTEMS PERFORMANCE SUMMARY

	Single-Family Residence	Multifamily Residence
System Features		
Collector area, sq ft	1000	10,000
Thermal energy storage tank, gal	1200	18,000
Heat pump capacity, Btu/hr	80,000	800,000
Auxiliary heater capacity, Btu/hr	80,000	800,000
Domestic hot water		
Storage tank, gal	50	650
Recovery rate, Btu/hr	60,000	750,000
Yearly Performance		
Residence load, 10^6 Btu	213	2130
Hot water load, 10^6 Btu	30.7	384
Solar contribution, %	61.2	63.4
Auxiliary thermal energy contribution, %	33.7	30.6
Auxiliary electrical energy contribution, %	5.1	6.0
Total energy expenditure		
Thermal (fuel oil or gas), 10^6 Btu	127	1100
Electrical, kw-hr	7670	87,900
Ultimate energy saving, 10^6 Btu/yr	162	1780
Present value benefit, $ (20-year, residential; 25-year, commercial)	3600	59,000

3. COLLECTOR SUBSYSTEM

3.1 GENERAL

A bid package was prepared by AiResearch and submitted to 31 collector manufacturers (see Table 3-1). The statement of work from this RFP is presented in Exhibit 3A at the end of this section. As evidenced by the list of Table 3-1, a variety of collector concepts will be evaluated. Major evaluation criteria will include the following:

(a) Overall design features and materials of construction

(b) Durability, potential corrosion, and insulation outgassing problems

(c) Warranty

(d) Performance over the range of $\Delta T/I$

(e) Collector capability to withstand repeated exposure to stagnation temperature

(f) Development status and availability of test data from actual installations

(g) Weight

(h) Production capability

(i) Manufacturer commitment to the solar energy field

(j) Collector cost as applicable to the present progam

(k) Estimated cost in volume production

The number of candidate collectors will be narrowed by evaluation of the data furnished in the proposals. Competing collectors will be further evaluated using a present value cost-effectiveness approach. The performance characteristics of these collectors will be used in AiResearch system computer

TABLE 3-1

PROSPECTIVE COLLECTOR MANUFACTURER BIDDERS LIST

AMETEK Hatfield, Pa.	Martin Marietta Denver, Colo.
Calmac Mfg. Corp. Englewood, N.J.	Northrop Inc. Hutchins, Tex.
Chamberlain Mfg Corp. Waterloo, Iowa	Owens Illinois Inc. Toledo, Ohio
Corning Glass Co. Corning, N.Y.	PPG Industries Pittsburg, Pa.
Daystar Corp. Burlington, Mass.	RAYPACK Westlake Village, Calif.
ENERGIX Corp. Las Vegas, Nev.	Refrigeration Research Inc. Solar Research Div. Brighton, Mich.
Energy Converters Inc. Chattanooga, Tenn.	Revere Copper and Brass Inc. Rome, N.Y.
Energy Systems Inc. El Cajon, Calif.	Reynolds Metals Co. Richmond, Va.
FMC Corp. Engineering Systems Div. Santa Clara, Calif.	SOLARAY Corp. Honolulu, Hawaii
Fun & Frolic Inc. Madison Heights, Mich.	Solar Energy Systems, Inc. Pennsauken, NJ
General Electric Valley Forge, Pa	Solar Systems Inc. Tyler, Tex.
Grumman Aircraft Corp. Long Island, N.Y.	Sunsource Inc. Beverly Hills, Calif.
Halstead-Mitchell Scottsboro, Ala.	Sun Systems Inc. Eureka, Ill.
International Environment Corp. Mamaroneck, N.Y.	U.S. Solar Corp. Silver Spring, Md.
KTA Corporation Rockville, Md.	Wolverine Division UOP Inc. Decatur, Ala.
Libby Owens Ford Toledo, Ohio	

program, and heating systems will be optimized to determine collector area requirement using present value cost as the optimization criterion. Collector sizing will be done through performance evaluation over a complete year of operation.

It is anticipated that actual site data will not be available at the time of collector evaluation. The residence models and Madison weather tapes furnished by NASA will be used as a basis for optimization of collector sizes and for comparison of the various collector approaches. Before a collector manufacturer is selected, it is suggested that a coordination meeting be held between AiResearch and NASA to review the complete evaluation procedure and data. It is felt that NASA through several years of active collector evaluation and in-house testing can offer AiResearch expertise that will be valuable in the final assessment of the relative merits of competing collector designs.

The collector subsystem approach described above is based on assumptions that have significant system implications. These are briefly discussed below.

The collector panels will require particular care to prevent corrosion during the more than 20-year life of the system. An interchanger is incorporated in the collector loop for this purpose. The penalties associated with use of the interchanger have been assessed (see Section 12) and found to be significant in terms of present value cost. These penalties will be considered in the evaluation of competing collector plate materials.

The collector panels can be exposed repeatedly to stagnation temperatures without damage. During the summer, collected solar energy is used for the sole purpose of domestic hot water heating. The energy storage tank is maintained at a maximum temperature of 160°F. When this temperature is achieved, the collector pump is deactivated and the collector is drained.

Should the collector design be such that it cannot be subjected to the high stagnation temperatures without damage, then the collector loop configuration must be modified to prevent such occurrences. Figure 3-1 shows the arrangement of a collector loop designed to assure flow through the collector at all times (when exposed to solar radiation). This loop features (1) redundant pumps with automatic switchover controls, (2) redundant collector control and sensors, and (3) means of dissipating the excess heat collected through an air cooler.

As an alternate approach, night radiation could be used for rejection of the excess solar energy collected during daytime. In either case, the collector loop required is significantly more complex and costly than the basic loop. This added complexity and the cost of the equipment will be considered in the evaluation of competing collectors.

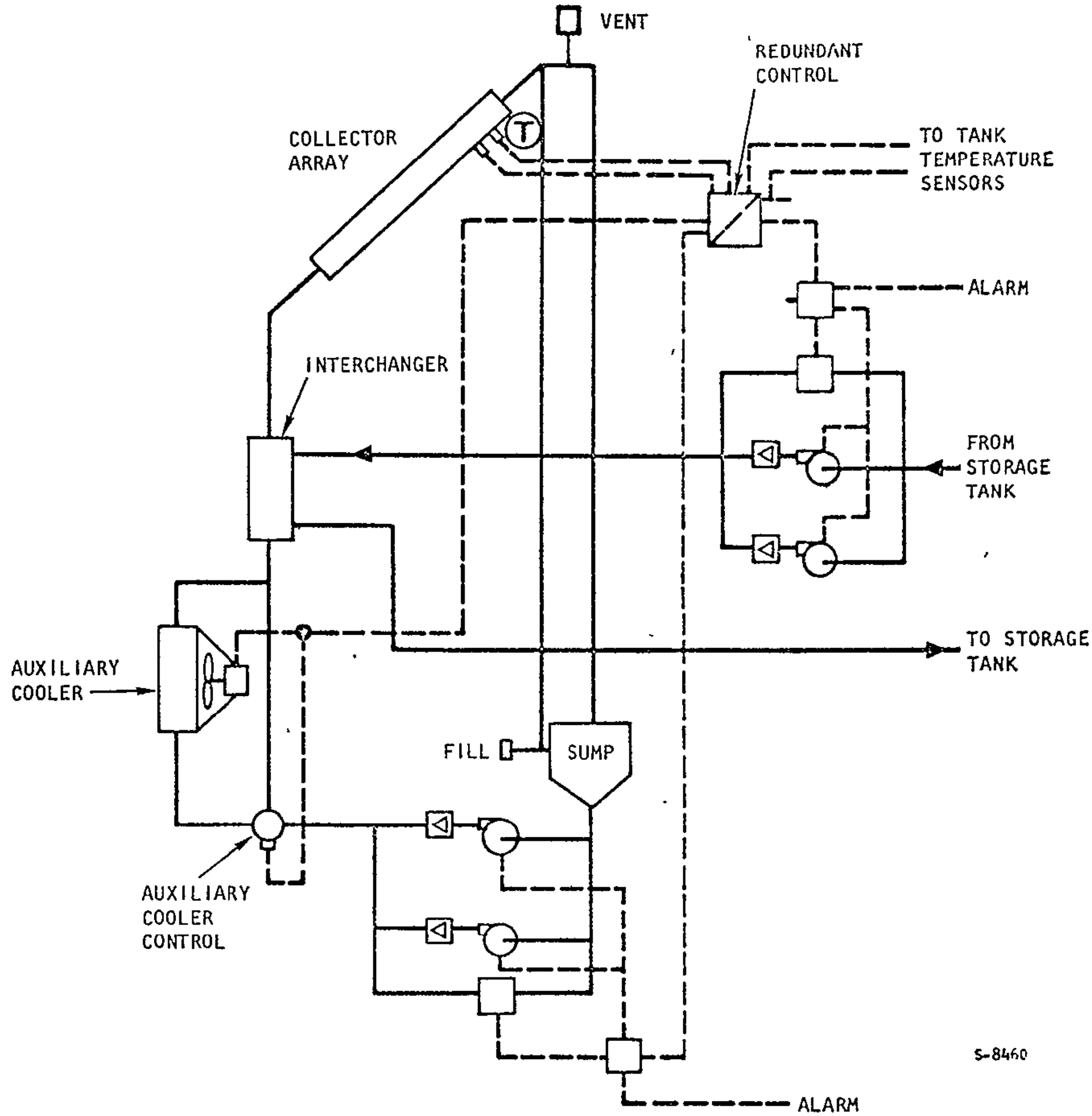


Figure 3-1. Candidate Collector Loop Configuration

EXHIBIT 3A

REQUEST FOR PROPOSAL

SOLAR COLLECTORS FOR HEATING
AND COOLING SYSTEMS

EXHIBIT 3A

REQUEST FOR PROPOSAL

SOLAR COLLECTORS FOR HEATING

AND COOLING SYSTEMS

INTRODUCTION

AiResearch Manufacturing Company of California has recently been awarded a contract by the National Aeronautics and Space Administration, Marshall Space Flight Center, for the design and development of solar heating and cooling systems.

Under the present contract, 12 systems of various sizes will be designed, fabricated, certified, and installed at various locations in the U.S. for long-term evaluation. Site selection is not definitized at this time.

This request for proposal covers engineering activities in support of the AiResearch program, and fabrication and delivery of solar collectors for the 12 systems. It is estimated that approximately 50,000 sq ft of collector will be procured. The exact number of collector modules will be defined prior to placing the purchase order.

<u>It is not the intent of this RFP to cover the development of collectors. AiResearch plans instead to procure collectors that are in an advanced development status, and for which test data are available to (1) substantiate performance and life, and (2) demonstrate the durability of the proposed design.</u>

It has been determined that only double glazed configurations will offer satisfactory performance. Within that limitation, as many collector configurations as desired may be proposed to include available options such as selective coatings. Technical and price data shall be submitted for each option proposed.

PROPOSAL INSTRUCTIONS

Proposals shall be submitted in three parts in a single volume.

Technical

Management

Cost

It is requested that six copies of the proposal be provided. The data required shall be furnished in as brief a manner as possible consistent with completeness. Collectors will be evaluated using the following criteria, which are not listed in order of importance.

(a) Design features

(b) Development status

(c) Collector life and maintainability

(d) Compliance with applicable portions of the NASA Solar Interim Performance Criteria

For all collectors proposed which appear to meet minimum standards of life, durability, and safety suitable for the intended application, a further evaluation of performance versus cost will be conducted. The collector representing the most cost effective design when operating with the AiResearch solar heating and cooling systems will be selected for procurement.

PROGRAM SCOPE

The program includes two major tasks, as follows.

Task I--Collector Certification

Certification of the collector by an independent and approved agency is required as part of this program. Certification shall include performance verification and detail evaluation of the design to demonstrate conformance to the NASA interim performance criteria. These criteria will be supplied to the proposed upon request.

Task 2--Fabrication and Preparation for Shipment

Under this task, collector modules shall be fabricated and packaged for shipment. The collector shall be a prefabricated unit consisting of absorber plate, fluid passages, transparent covers, and frame with insulation. Assembly of the collector panels into a complete array is not part of this contract. However, any hardware developed for the purpose of collector installation, including support frames, manifolds, and connecting lines, should be described for procurement at a later date. Cost data on these items should also be furnished separately.

It is currently estimated that about 50,000 sq ft of collector panels (effective area) will be required. System optimization studies to be performed following site selection will determine the exact number of collector panels to be procured under this contract.

TECHNICAL DATA REQUIREMENTS

Proposals shall contain the following information:

(1) Description of collector modules

(2) Materials of construction

(3) Drawings or sketches in sufficient detail for assessment of the drawing

(4) Definition of installation interfaces and supports

(5) Recommendations for corrosion inhibitors

(6) Installation features

(7) Maintenance features--scheduled maintenance requirements, if any

(8) Estimated life--provides available life data

(9) Performance in the form of efficiency versus $\Delta T/I$--define collector area used in efficiency calculation at ΔT terms.

All data necessary for performance modeling of the collector in actual installation shall be provided. These data shall include, as a minimum,

(a) Glazing reflectance as function of sun angle

(b) Effect of fouling a performance

(c) Performance degradation due to sand abrasion

(d) Pressure drop characteristics and recommended flow rate

(10) Stagnation temperature data including frame temperatures

(11) Capability of collector to withstand repeated exposure to stagnation temperature

(12) Effective collector area

(13) Weight

(14) Estimated installation cost data based on past installations

(15) Durability and performance degradation

(16) Draining features

(17) Weight packaged for shipment

DELIVERY REQUIREMENTS

It is anticipated that contract award will be November 1, 1976. Delivery of 2000 sq ft will be required November 1, 1977. The remainder are to be delivered February 1, 1978. Certification of the collectors must be completed prior to delivery and acceptance of the panels.

MANAGEMENT DATA REQUIREMENTS

The following data shall be supplied:

(1) Production capability

(2) Experience--past production, current programs, existing installations, related products.

(3) Quality assurance methods to assure product repeatability

(4) Product warranty

COST DATA REQUIREMENTS

A firm fixed-price quotation for each of the tasks described above shall be submitted. Price data shall be furnished for each task separately.

Price data shall be supplied for all alternate collector configurations proposed.

The final order will be in the range of 40,000 to 60,000 sq ft. Proposer is to stipulate if there is a price difference per square foot within that order range.

Prices should be FOB sellers plant.

4. THERMAL ENERGY STORAGE

4.1 GENERAL

Water was selected as the thermal energy storage medium. The design of the water tank itself does not present any problem. System analyses performed prior to the proposal were conducted on the assumption that the tank was completely mixed. This constitutes a conservative approach.

Significant performance improvements could be realized through thermal stratification of the tank. Collector feed water temperature would be minimized, and the water temperature at heat pump inlet would be increased. To realize these advantages, a study was conducted to develop a simple tank manifold design that would promote tank stratification. The results of these studies are presented below.

Verification testing of this manifolding approach will be conducted using a simple tank model constructed of clear plastic for visual observation.

4.2 MANIFOLD DESIGN

The tank model used for manifold investigations has a capacity of about 1200 gal. The tank is cylindrical and positioned vertically; overall dimensions are 5 ft in dia by 8 ft long. Flow rates through the interchanger and heat pump circuits are 30 and 15 gpm, respectively. This tank and associated flows are representative of a single-family residence system (see Figure 4-1).

The manifold was designed to minimize turbulence and secondary current formation inside the tank in the presence of incoming or outgoing flows. The design concept is illustrated in Figure 4-2. The manifold features concentric

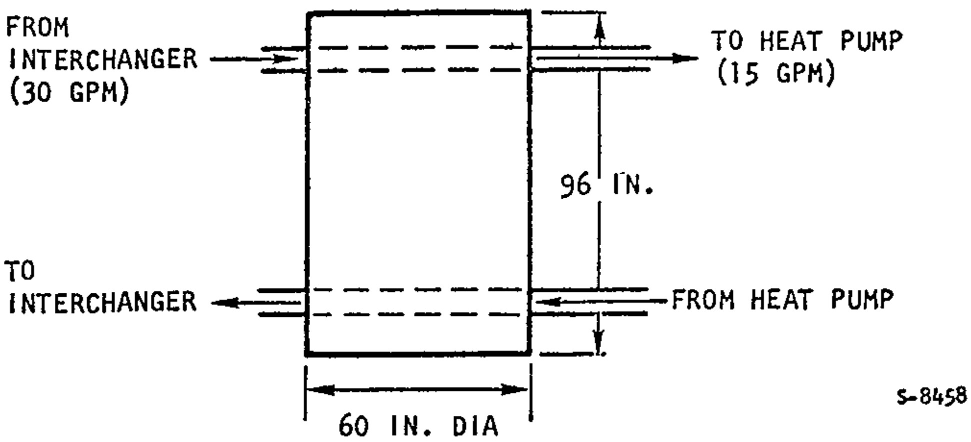


Figure 4-1. Single-Family Residence Water Tank

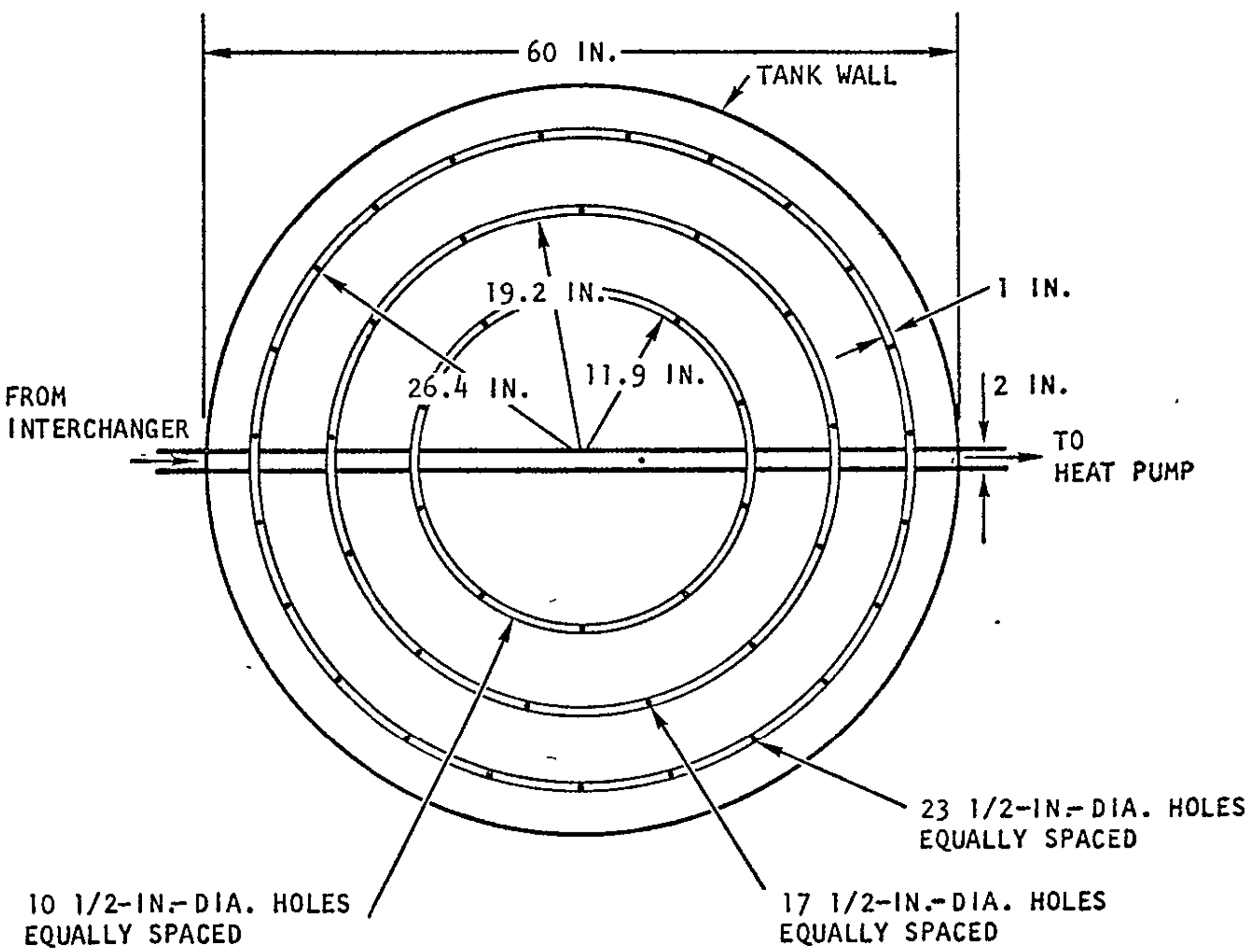


Figure 4-2. Typical Manifold Design

lines with holes equally spaced along the lines. The water feed is from a straight section of pipes across the tank. The holes discharge the water upward.

During solar energy collection, hot water from the collector loop is directly available to the heat pump through the straight section of pipe. The model shown features three rings with a total of 50 1/2-in.-dia holes equally spaced. This manifold would be installed 2 in. below the surface of the water.

Any number of rings and holes can be used; several schedules of numbers of holes, hole diameters, and depth of installation have been estimated. They are listed in Table 4-1.

In practice, the actual number of rings/holes will be determined by fabrication cost. It may be that the larger number of rings constitutes the less expensive approach since the rings can be fabricated easily of small size (less than 1-in.-dia) tubing. It should be noted that the larger number of rings also represents the best solution since it is more representative of a screen that would be installed across the tank.

4.2.1 Design Verification Testing

Demonstration of the performance of the manifold approach described above is planned. A clear plastic tank will be used. Tank and ring dimensions are shown in Figure 4-3. The test will be conducted as follows. The flow in the tank will be stabilized, then a dye will be injected into the tank feed streams and the flow pattern will be observed from two directions across the tank.

The test rig is designed so that the manifold can be modified easily to change the hole distribution around the circumference of the rings, the hole size, or the depth of the manifold below the surface of the water. Other manifold configurations with 3 or 4 rings also could be tested if the 2-ring configuration is not satisfactory.

TABLE 4-1

CANDIDATE MANIFOLD DESIGNS

Number of Holes	Hole dia, in.	Number of Rings	Manifold Depth Below Surface, in.	Ring radius, in. (number of holes per ring)						
				Ring No. 1	Ring No. 2	Ring No. 3	Ring No. 4	Ring No. 5	Ring No. 6	Ring No. 7
196	1/4	7	1	5.49 (9)	9.26 (15)	13.03 (22)	16.80 (28)	20.57 (34)	24.34 (40)	28.12 (48)
88	3/8	5	1-1/2	4.53 (5)	10.19 (11)	15.85 (18)	21.51 (24)	27.17 (30)	-	-
50	1/2	3	2	11.94 (10)	19.16 (17)	26.39 (23)	-	-	-	-
32	5/8	2	2-1/2	17.31 (13)	25.77 (19)	-	-	-	-	-

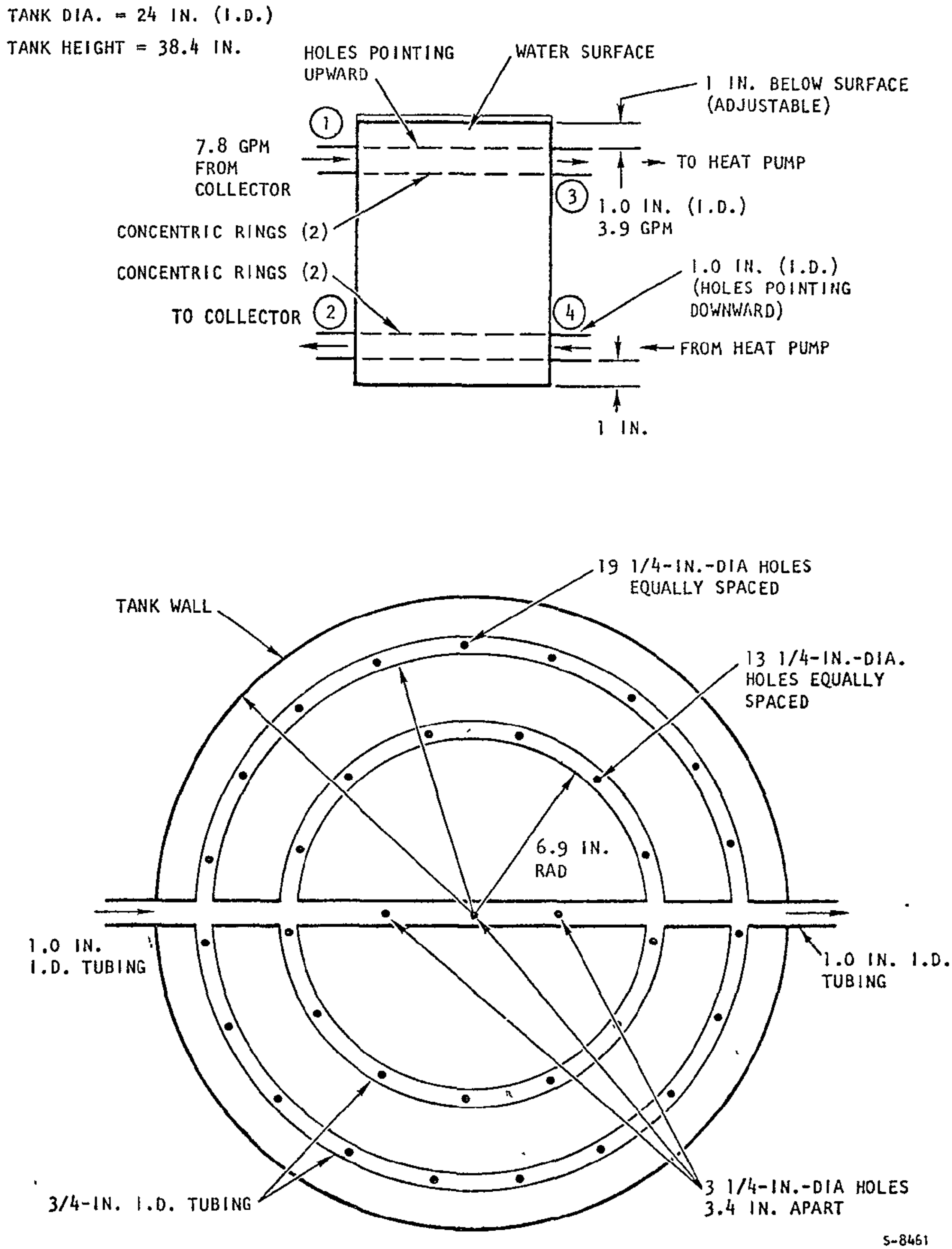


Figure 4-3. Tank/Manifold Configuration for Stratification Test

Testing will be conducted to simulate the various operating conditions of the tank with respect to collector/heat pump flows. The flows listed below scaled from the full-size tank will be used in the test.

Tank Inlet/outlet Station (see Figure 4-3)

Condition	1	2	3	4
1	7.8 gpm	7.8 gpm	-	-
2	-	-	3.9 gpm	3.9 gpm
3	7.8 gpm	7.8 gpm	3.9 gpm	3.9 gpm

5. SPACE HEATING SUBSYSTEM

5.1 GENERAL

The preliminary design of the heat pump in both the heating and cooling modes of operation has been completed. Subsystem performance characteristics were determined and problem statements for all heat pump components were released. These problem statements cover the heating and cooling cases.

Data presented below include subsystem-level cooling mode data since these are essential to the design of the heat pump package. The heat pump rated capacities listed below were selected for the application shown.

	Heating	Cooling
Single-family residence	60 KBTUH	3 tons
Multifamily residence	600 KBTUH	25 tons
Commercial application	200 KBTUH	10 tons

Note that the heating capacity of the overall system could be higher than shown if desired through the use of larger auxiliary heaters. System-level trade studies have shown that the most economical system design does not feature a collector sized to satisfy the maximum heating demand. It follows that the heat pump capacity for a given installation is collector-limited.

5.2 60-KBTUH/3-TON HEAT PUMP

5.2.1 Heating Mode Operation

5.2.1.1 Heat Pump Description and Control

A schematic of the 60,000-Btu/hr capacity heat pump is shown in Figure 5-1. The cycle has been described previously (see Section 2).

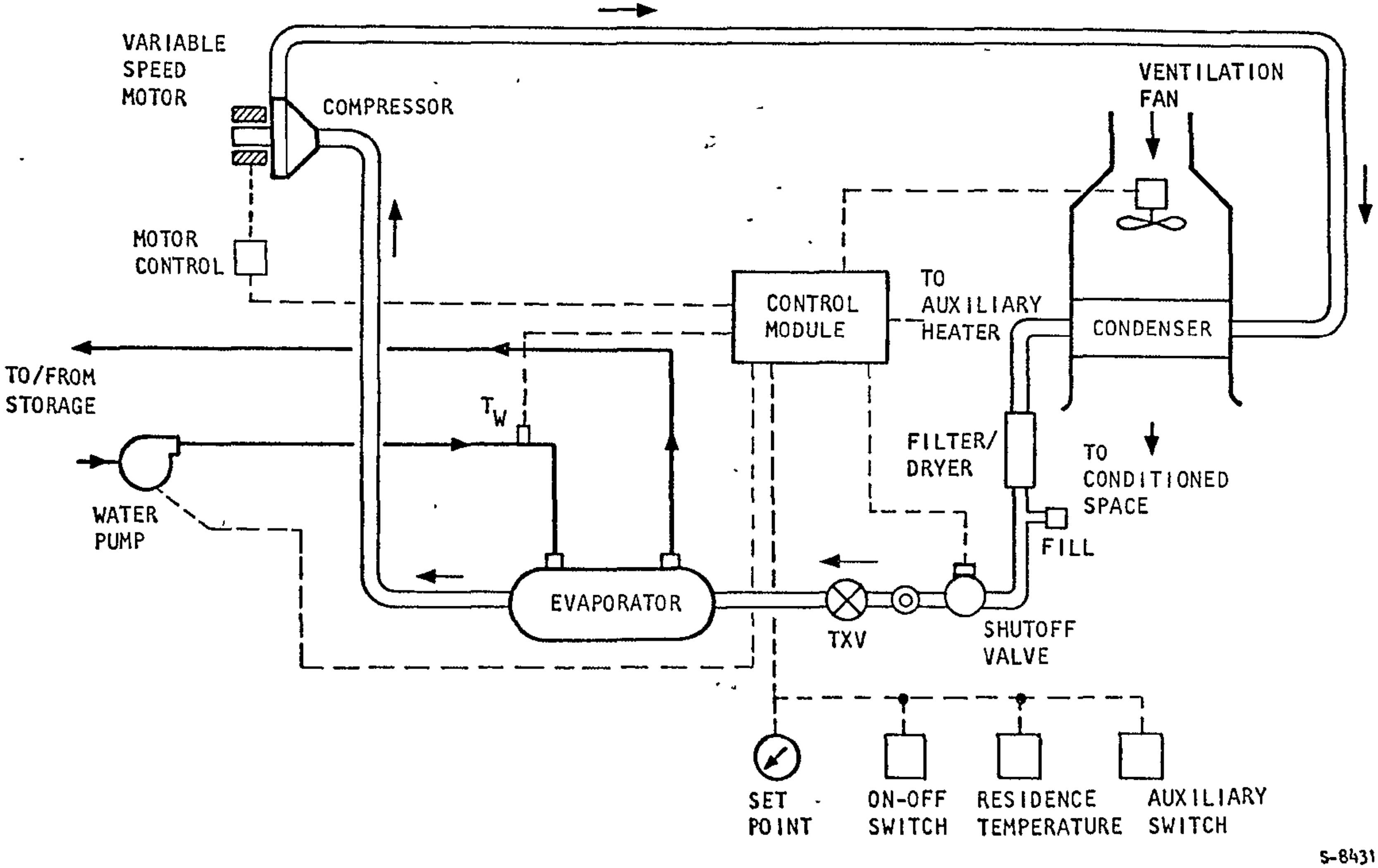


Figure 5-1. Single-Family Residence Space Heating Subsystem

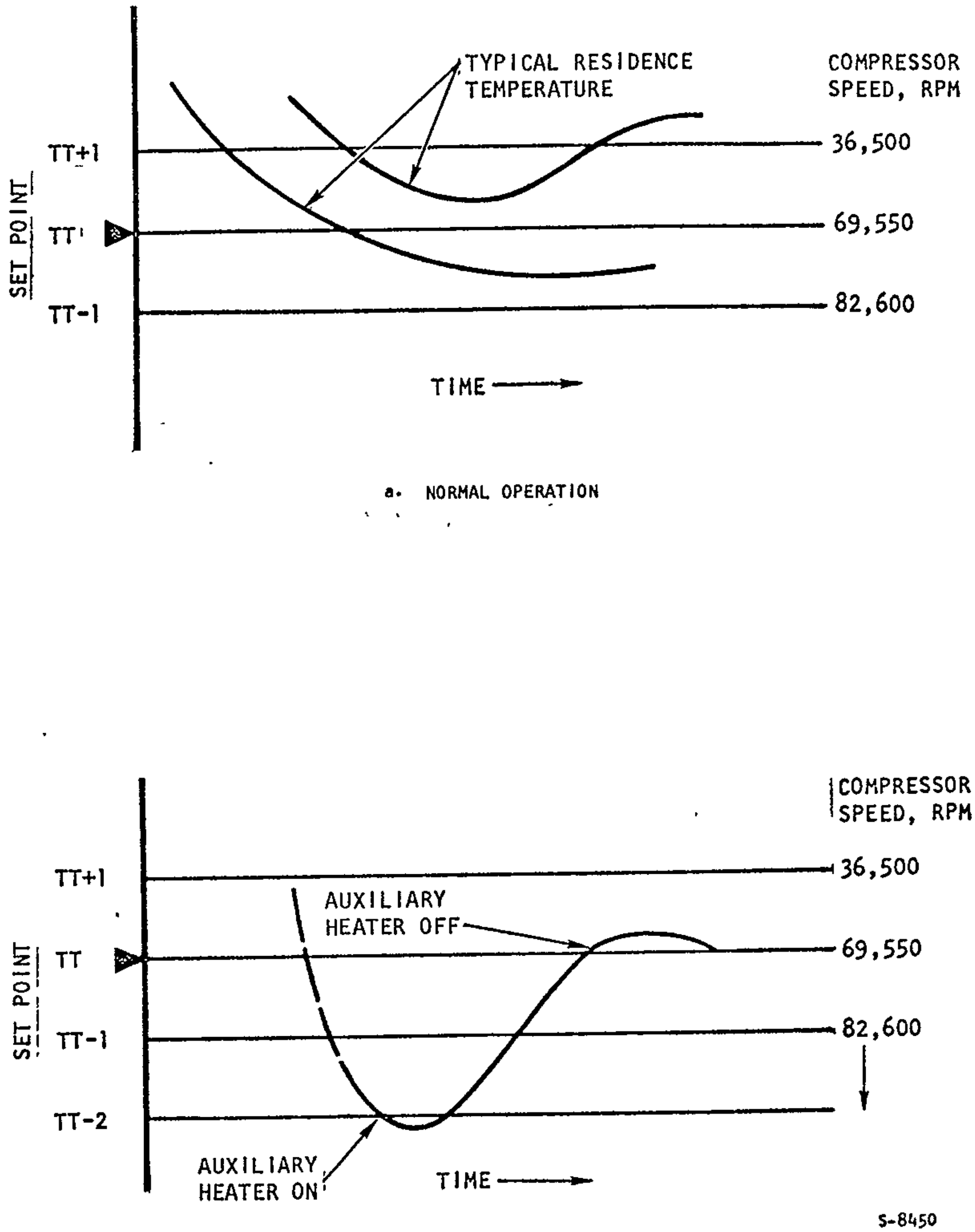
The following signals and switches are used for control: (a) temperature set point, (b) tank water temperature, (c) residence temperature, (d) ON-OFF switch, and (e) auxiliary heater switch. Using the above parameters, a control scheme was developed whereby compressor speed is adjusted so the residence temperature is maintained within 2°F (±1°F) of the set point under any normal operating condition.

The control sequence is illustrated in Figure 5-2. The heat pump will start when the residence temperature, TR, is 1°F above the desired set point, TT. This will involve switching on the water pump, the ventilation fan, and the compressor motor. The compressor motor will start at minimum speed, 36,500 rpm. Should the residence temperature drop, the compressor speed will be commanded to a higher value, thus increasing the capacity of the heat pump. Maximum compressor speed and capacity will be reached when the residence temperature is 1°F below the set point.

During normal operation, a compressor speed will be reached corresponding to a heat pump capacity which maintains the residence within the 1°F band on either side of the set point.

Should the capacity of the heat pump at maximum speed be lower than the residence load, the residence temperature will drop further and the auxiliary heater will be switched on when TR = TT - 2°F.

The capacity of the auxiliary heater is not controlled; it always delivers at full capacity. The total system capacity, therefore, is that of the auxiliary heater plus that of the heat pump. Under these conditions, the residence temperature will increase and the auxiliary heater will be switched off when TR = TT. Heat pump speed will then be 69,500 rpm. The heat pump will continue to operate and (assuming that the residence loads have not changed) the residence temperature will drop. The entire cycle will be repeated.



a. NORMAL OPERATION

b. MAXIMUM LOAD OPERATION (LOAD > CAPACITY)

Figure 5-2. 60 KBTUH Heat Pump Control

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Note that under this extreme condition the control band is also 2°F from TT to TT - 2°F.

The capacity of the heat pump is also related to the temperature of the water in the thermal energy storage tank so that operation in the "maximum load" mode could be due to low water temperatures rather than high residence loads. In general, low water temperature and high residence loads will occur at the same time because of (1) low insolation and (2) high solar energy utilization.

The minimum capacity of the heat pump is determined by the speed, flow, and adiabatic head characteristics of the compressor. The compressor map is shown in Figure 5-3; superimposed on the compressor map are the heat pump operating characteristics in terms of capacity and water source temperature. This map represents a compromise between heating mode and cooling mode operating range. If the compressor were designed for heating only, a wider range of capacity could easily be achieved.

The surge characteristics of the compressor must be incorporated into the control system to assure stable operation. In the heating mode of operation, the minimum compressor speed can be defined as a function of the water heat source temperature. This relationship is shown in Figure 5-4, which illustrates the operating range of the heat pump.

A water temperature sensor at heat pump inlet is used to provide the control module with the information necessary to maintain compressor speed above the compressor surge limit.

Section 8 presents the details of the control module preliminary design.

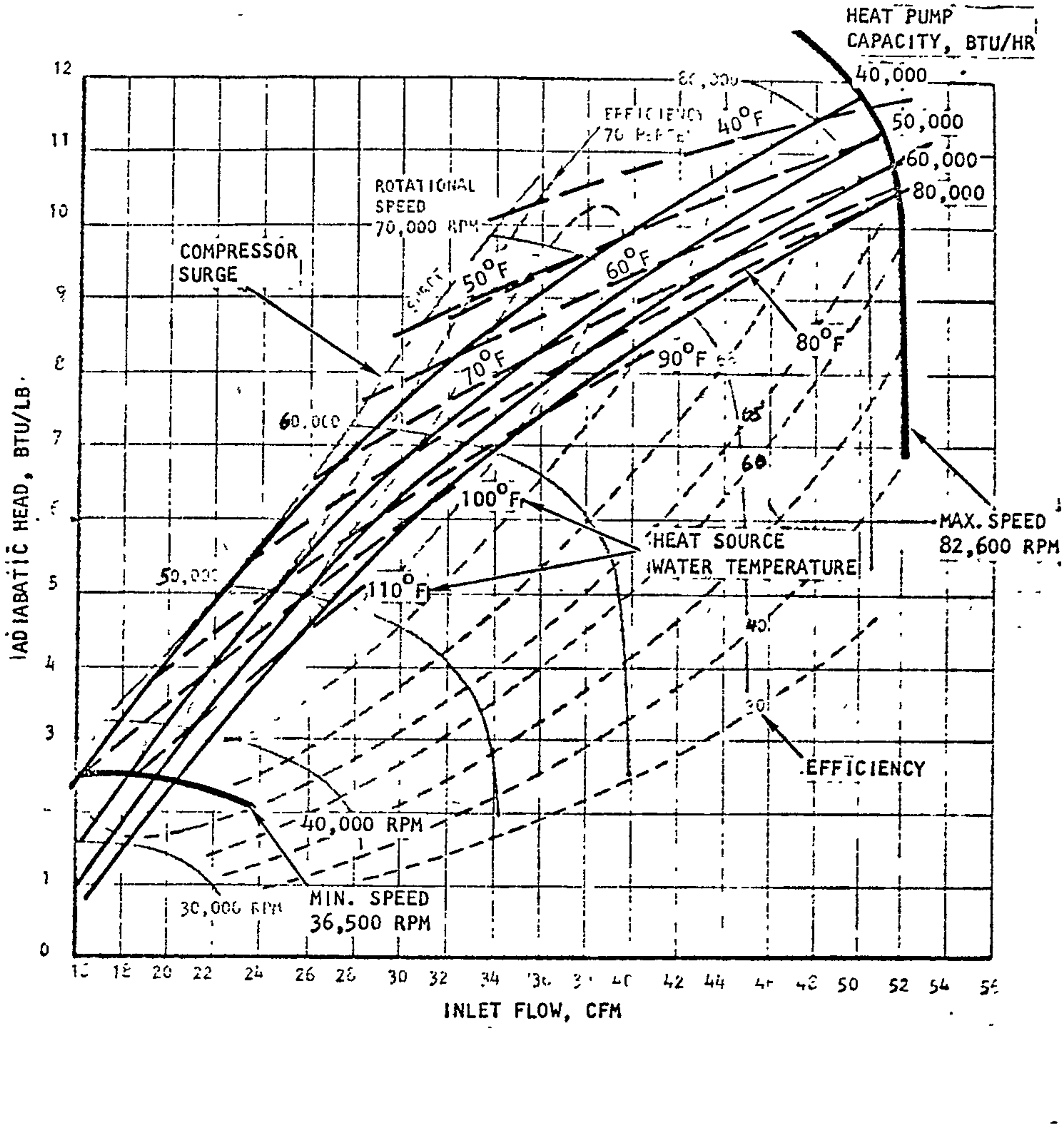

Figure 5-3. 60 KBTUH Heat Pump Compressor in Heating Mode

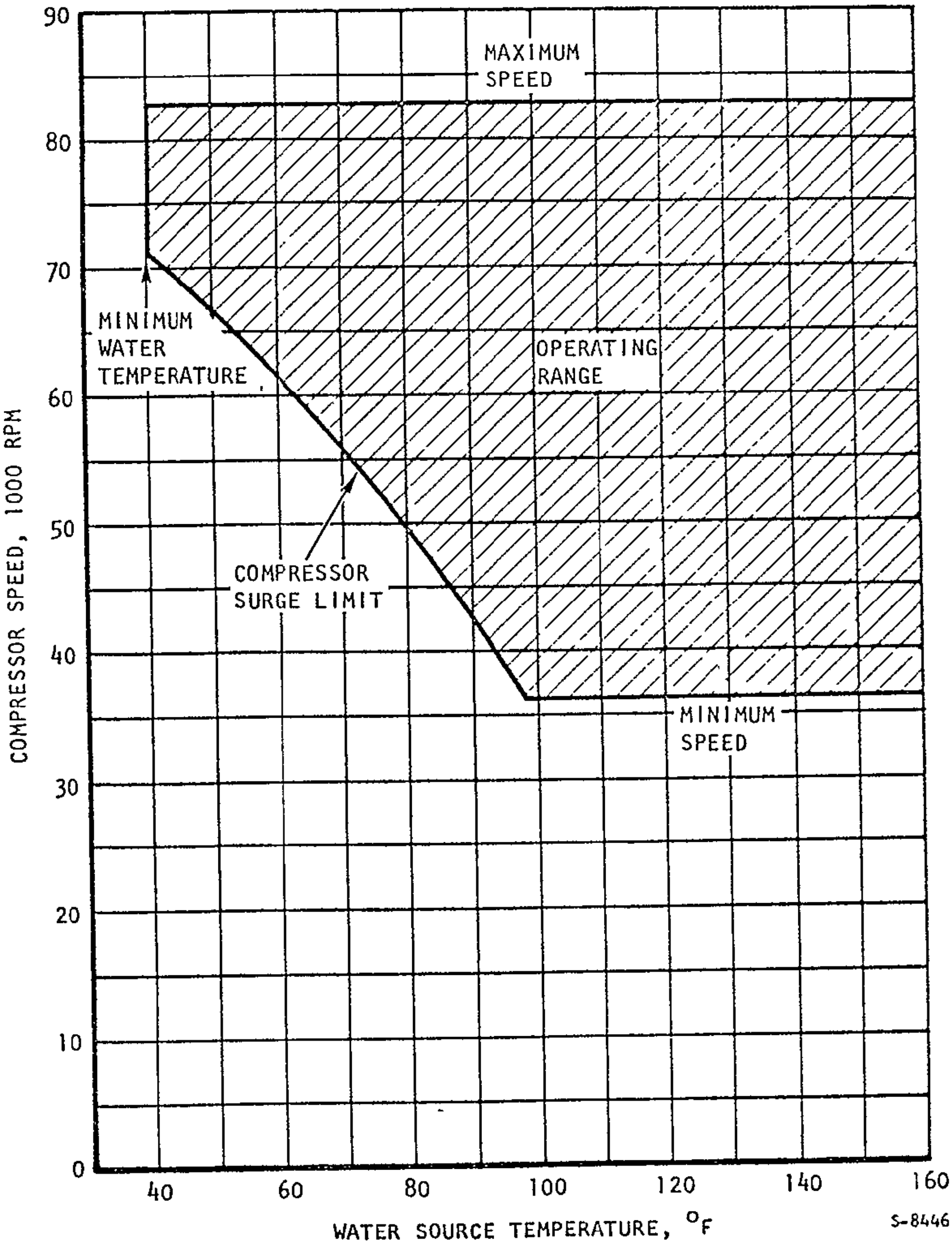

Figure 5-4. 60 KBTUH Heat Pump Operating Range

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5.2.1.2 60 KBTUH Heat Pump Performance

The heat pump is designed to provide 60,000 Btu/hr heating capacity with a water heat source temperature of 60°F and a residence temperature of 70°F (standard ARI rating conditions). As mentioned previously, the compressor was designed to provide a wide operational range in both the heating and cooling modes of operation; in this manner the same compressor, motor, and motor control can be used for the heating-only and for the heating-cooling systems. This is consistent with normal production planning for the consumer market.

Similarly, the design of the heat exchangers must accommodate the heating and cooling modes. This was the approach followed. Heat exchanger thermal and pressure drop performance was carefully balanced so that the same units could be used for both systems.

Figure 5-5 shows the thermodynamic characteristics of the heat pump at design point. These data were used in the design of the compressor, motor, and heat exchangers. Component data are presented later in this document.

The coefficient of performance of the 60 KBTUH heat pump is given in Figure 5-6 as a function of water source temperature and heat pump capacity. The COP plotted is defined as

$$COP = \frac{\text{HEATING EFFECT}}{\text{TOTAL ELECTRICAL POWER INPUT}}.$$

where the electrical power input includes the following:

(a) Compressor motor

(b) Ventilation fan

(c) Water pump

(d) Controls

COP at maximum speed is about 4.0, which compares very favorably with commercially available water source heat pumps.

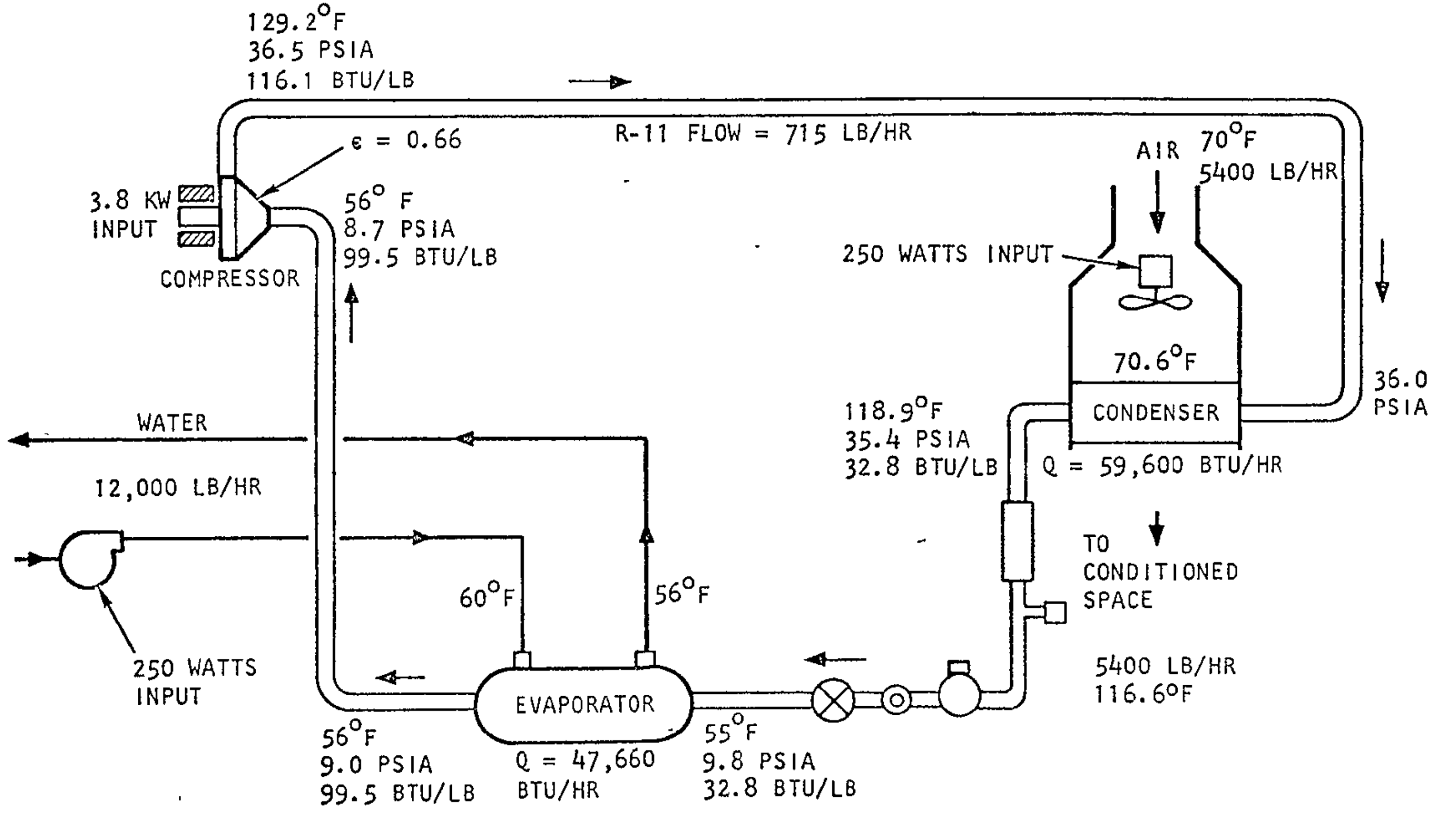

Figure 5-5. 60 KBTUH Heating Subsystem--Design Point Performance

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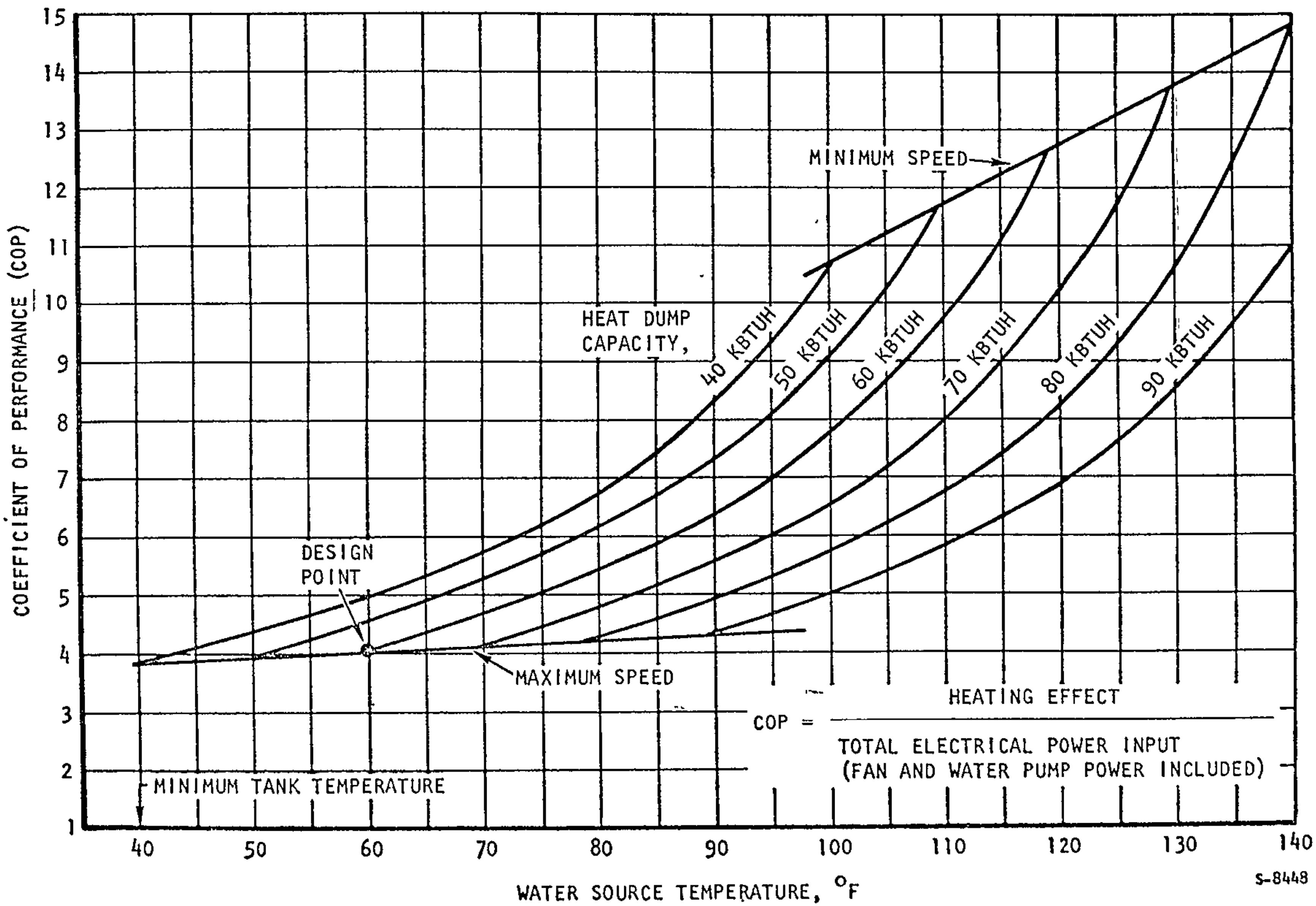

Figure 5-6. 60 KBTUH Heat Pump Coefficient of Performance

5.2.2 Cooling Mode Operation

The data presented on the cooling mode operation are only sufficient for a basic understanding of the design of the heat pump and its components. More data on the heat pump in the cooling mode of operation will be presented at the time of cooling system PDR.

5.2.2.1 Heat Pump Description and Control

Figure 5-7 is a schematic of this subsystem. The functions of two system heat exchangers are reversed from the heating to the cooling mode. The heat pump effect heat exchanger always uses the conditioned space air as a heat exchange fluid. The other uses either cooling tower water as a heat sink (in the cooling mode) or hot water from the thermal energy storage tank as a heat source (in the heating mode).

Six selector valves (one R-11 and five water), actuated from a mode selector switch within the residence, permit operation in the heating or the cooling mode. In the heating mode, operation is identical to that for the single-family residence heating subsystem described previously. In this mode, the compressor is motor-driven at variable speed and the turbine is inactive. The entire Rankine power loop is disabled by interruption of the heat input to the boiler and isolation of the cooling tower.

In the cooling mode of operation, heat energy from the hot water storage tank is used as the Rankine power loop heat source. A cooling tower serves as the ultimate heat sink. When thermal energy available from the storage tank is inadequate, the motor integral with the trubomachine is activated. In this augmented mode of operation, turbocompressor speed is controlled at a constant value by the motor, but the turbine still supplies a large portion of the power

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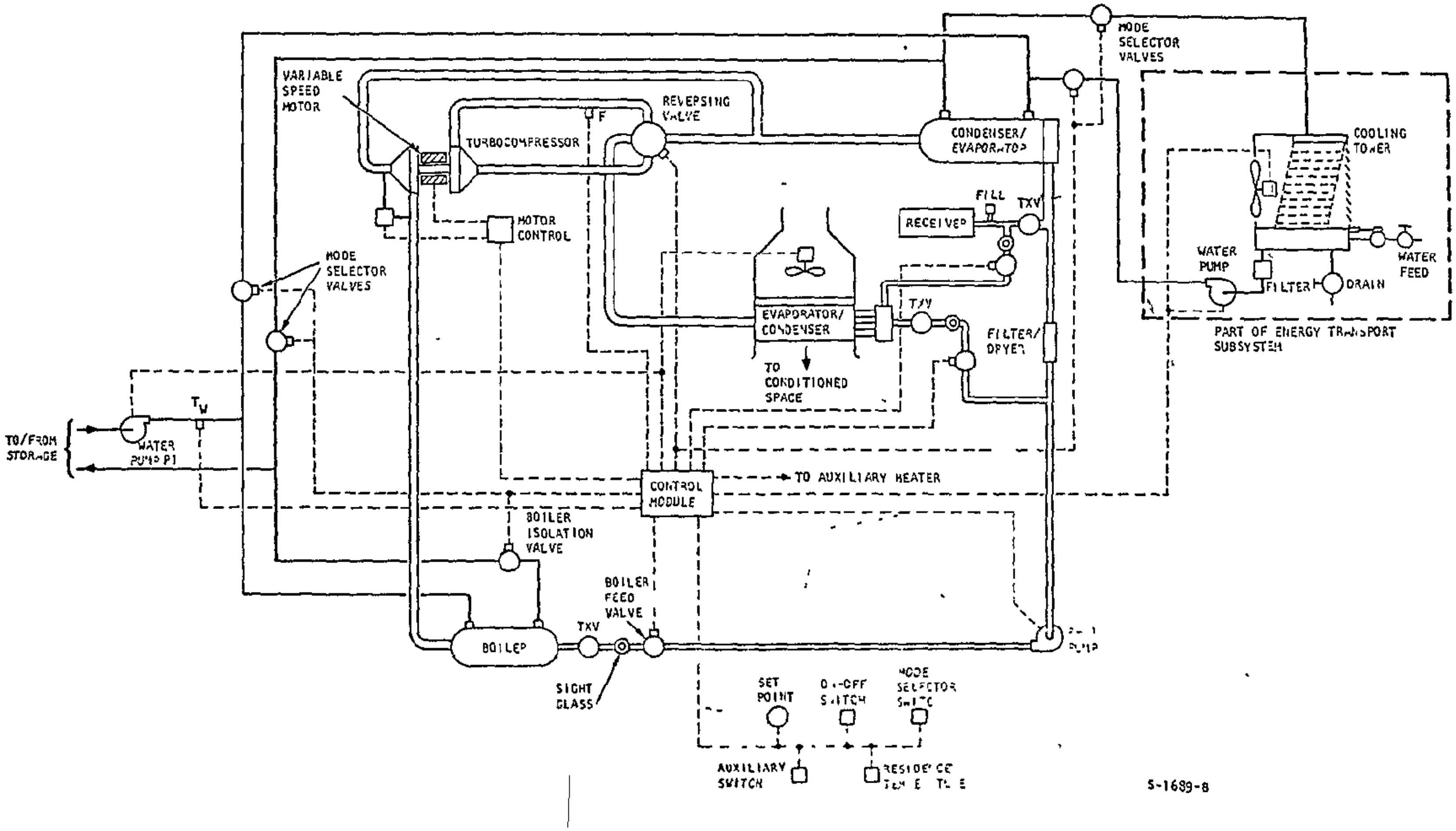

Figure 5-7. 3-Ton/60 KBTUH Cooling and Heating Subsystem

to drive the compressor. As the temperature of the water heat source to the boiler drops further to 145°F, the boiler is isolated and all power necessary to drive the compressor is developed by the electric motor.

The only residence control necessary for operation of the subsystem in the cooling mode, in addition to the heating mode sensors and controls described previously, is the mode selector switch. This switch will actuate (a) the three mode selector valves in the hot water supply lines, (b) the refrigerant reversing valve, (c) the cooling tower isolation valves, and (d) the boiler feed valve. In addition, the switch will control power to the R-11 pump and the cooling tower fan and water pump.

In the heating mode, the system is controlled as explained previously. In the cooling mode, the system is essentially controlled by the residence temperature set point, which activates the system when the temperature exceeds 1°F above the set point and deactivates it when the temperature drops 1°F below the set point.

When the air conditioner is activated from the temperature error signal, the control module will activate (a) the cooling tower fan and water pump, (b) the hot water pump, (c) the R-11 pump, and (d) the ventilation fan.

In normal operation, turbine power is sufficient to drive the compressor; the motor integral with the turbocompressor is inactive. As the water temperature from the storage tank drops, compressor speed is reduced, resulting in a loss of capacity. If the water temperature continues to drop, the power developed by the turbine will become inadequate. Turbine pressure ratio and speed are used to determine the condition at which the motor will be powered; the turbocompressor speed is then maintained constant at 66,000 rpm.

With the motor on, the turbine continues to provide a significant portion of the total power required to drive the compressor. When the water temperature decreases to 145°F, the control module closes the power loop, and the motor carries the entire load. A temperature sensor at the water pump outlet provides the signal. Power loop shutdown involves (a) R-11 pump deactivation, (b) water pump deactivation, and (c) closing the boiler isolation valve.

System shutdown from the normal operating mode (without motor augmentation) involves steps exactly the reverse of those listed above. With the motor on, system shutdown also involves deactivating the motor. The control module then is reset for operation in the normal mode. Shutdown occurs when the residence temperature drops 1°F below the residence temperature set point.

The operating range of the heat pump in the cooling mode is shown in Figure 5-8. The data are shown for a residence wetbulb temperature of 67°F (ARI rating condition). The plot shows a wide range of operation in terms of water source temperature and ambient wetbulb temperature.

The surge characteristics of the turbine-driven compressor are such that the compressor outlet volumetric flow is constant at any operational point on the surge line. This parameter was selected to provide the signal necessary to prevent compressor surge.

5.2.2.2 <u>Subsystem Performance</u>

The thermodynamic characteristics of the subsystem in the cooling mode of operation are shown in Figure 5-9. These data, together with the information presented in Figure 5-5, were used for equipment design.

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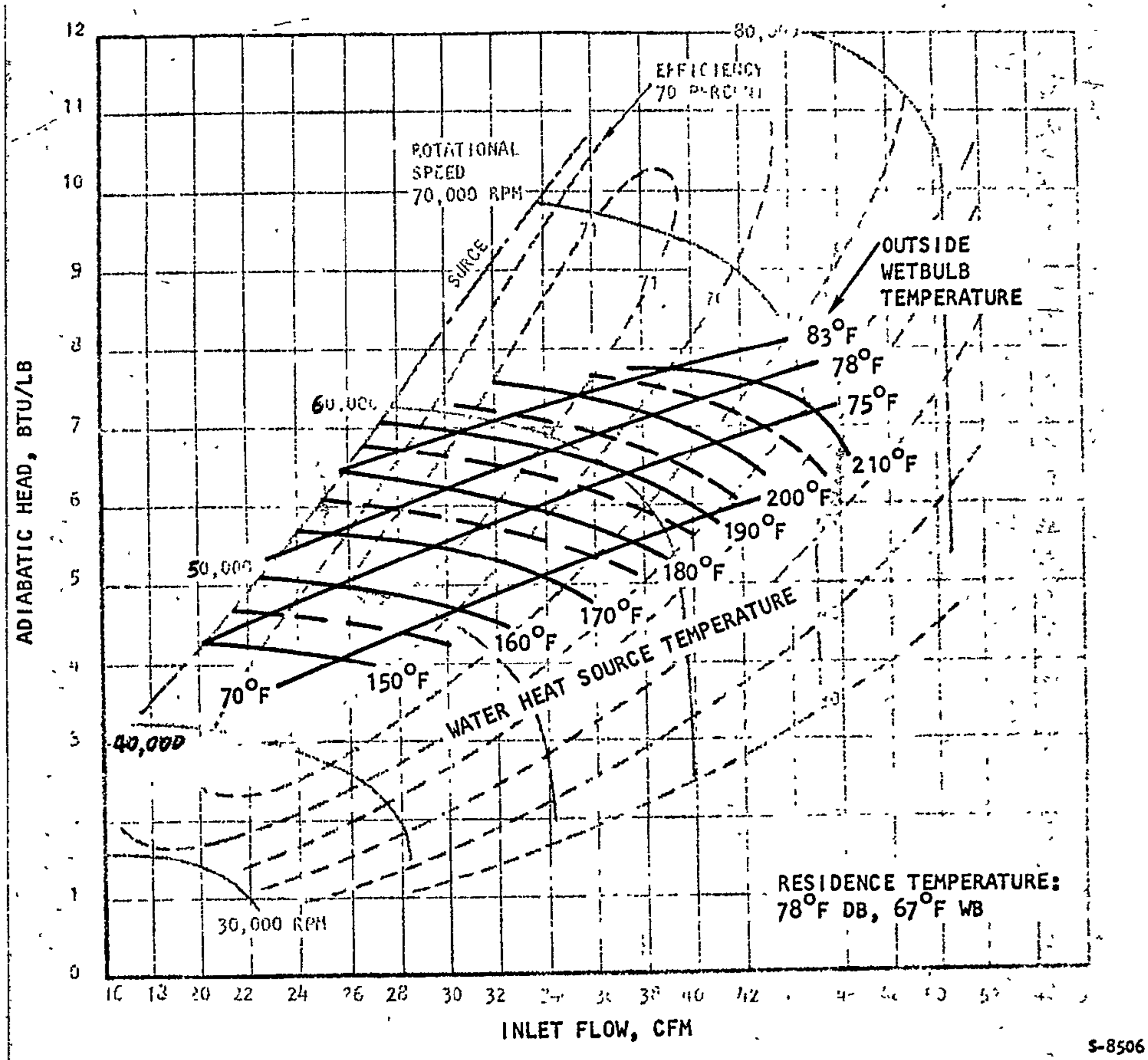

Figure 5-8. 3-Ton Heat Pump Compressor in Cooling Mode

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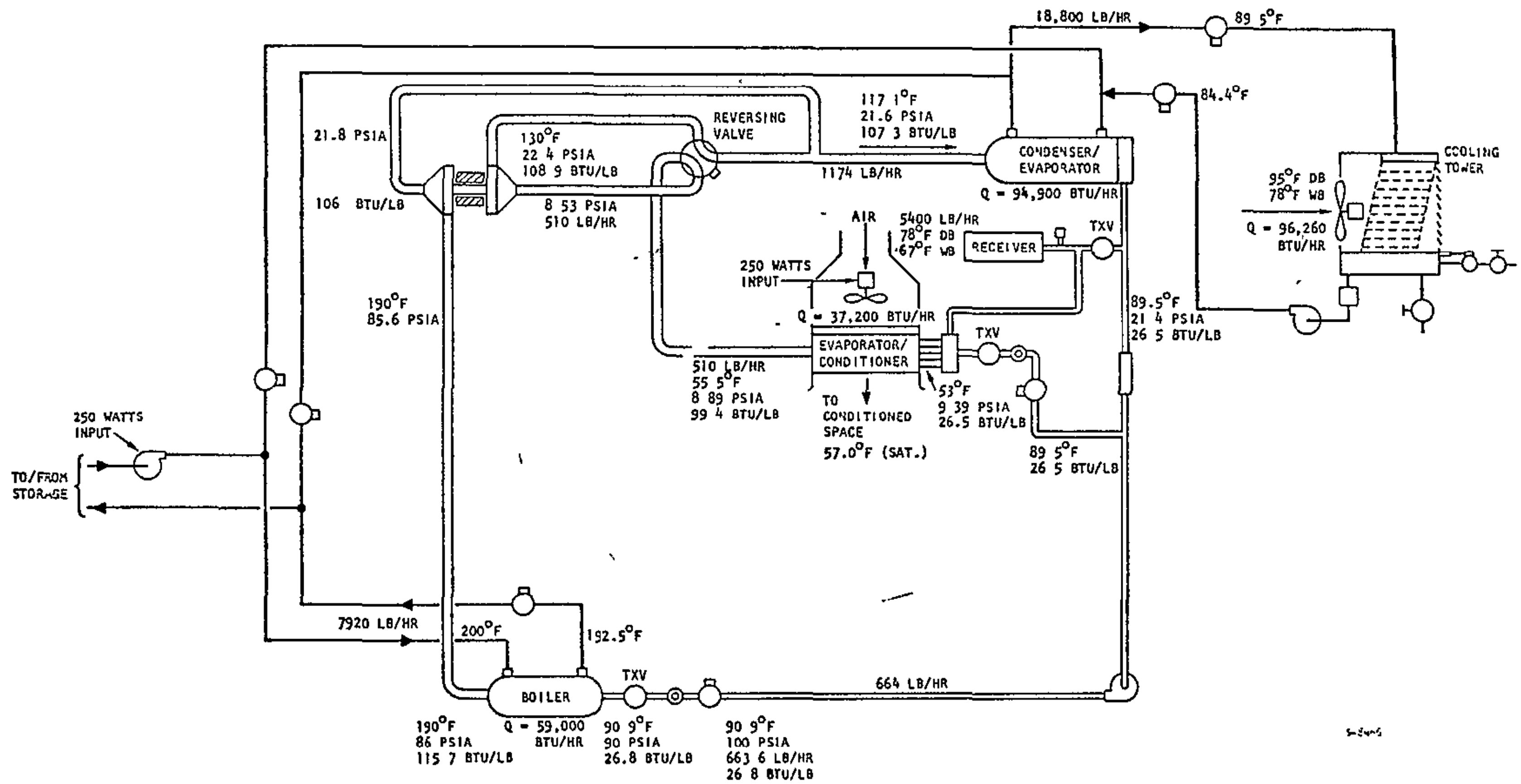

Figure 5-9. 3-Ton Cooling Subsystem--Design Point Performance

5.2.3 Heat Pump Package

Attached to this document is a layout of the 100,000-BTUH heat pump module. The package incorporates all heat pump equipment, the auxiliary heater, the solar collector loop pumps and interchanger, and the system control.

The package was developed to accommodate both the heating and heating/cooling systems. All duct connections are on the same side of the unit to facilitate through-wall or roof installation. Easy access is provided to all equipment that might require maintenance or replacement.

The overall dimensions of the package are 54 in. x 60 in. x 26 in. The height of the package is less than 30 in. to permit movement through normal doors.

The furnace heat exchanger and burner have a capacity of 100,000 BTUH. These assemblies are certified equipment from off-the-shelf furnaces repackaged for integration with the rooftop unit shown.

5.3 200 KBTUH/10-TON HEAT PUMP

This heat pump is identical in concept to the 60 KBTUH/3-ton unit described in para. 5.2.1.2. The schematic and the control logic are the same; however, the detailed characteristics of the subsystem differ. These characteristics are presented below in the same format used for the single-family residence unit.

5.3.1 Heating Mode Operation

5.3.1.1 Heat Pump Description and Control

Figure 5-1 shows the heat pump schematic. The control in the heating mode is identical to that described in para. 5.2.1.1. Compressor speed in this case is controlled between 57,440 and 27,500 rpm.

System operating characteristics superimposed on the compressor map are shown in Figure 5-10. The operating range of the heat pump within the compressor speed and surge limit is shown in Figure 5-11.

5.3.1.2 200 KBTUH Heat Pump Performance

The heat pump has a 200,000-Btu/hr capacity at design point corresponding to ARI standard rating conditions (water temperature 60°F, residence temperature 70°F). Thermodynamic characteristics of the heat pump loop at design point are shown in Figure 5-12.

The heat pump coefficient of performance for the range of water temperatures and capacities is shown plotted in Figure 5-13.

5.3.2 Cooling Mode Operation

5.3.2.1 Heat Pump Description and Control

A schematic of the 3-ton/60-KBTUH cooling and heating subsystem is presented in Figure 5-7. The unit is discussed in para. 5.2.2.1.

Figure 5-14 shows the operating range of the compressor in the cooling mode.

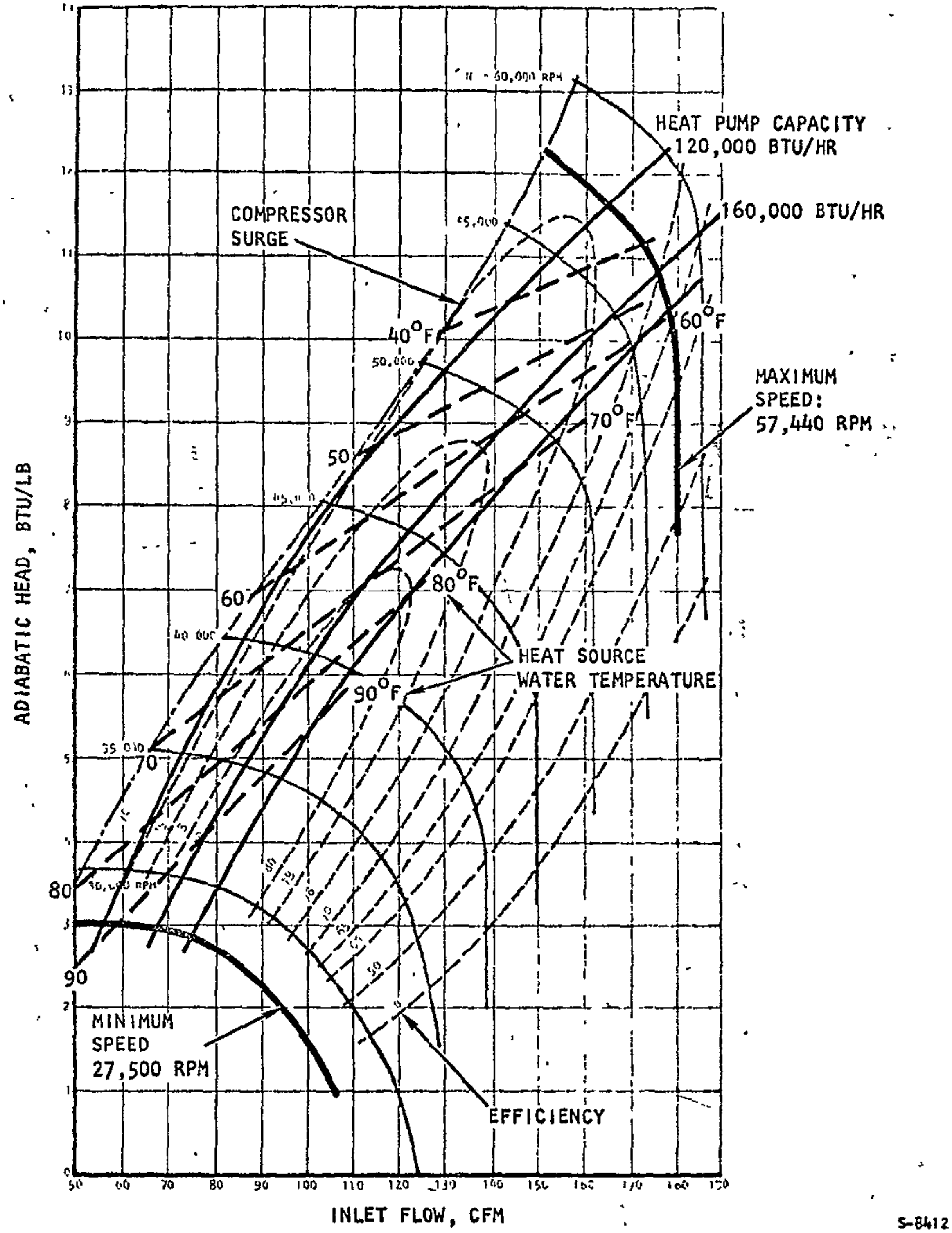

Figure 5-10. 200 KBTUH Heat Pump Compressor in Heating Mode

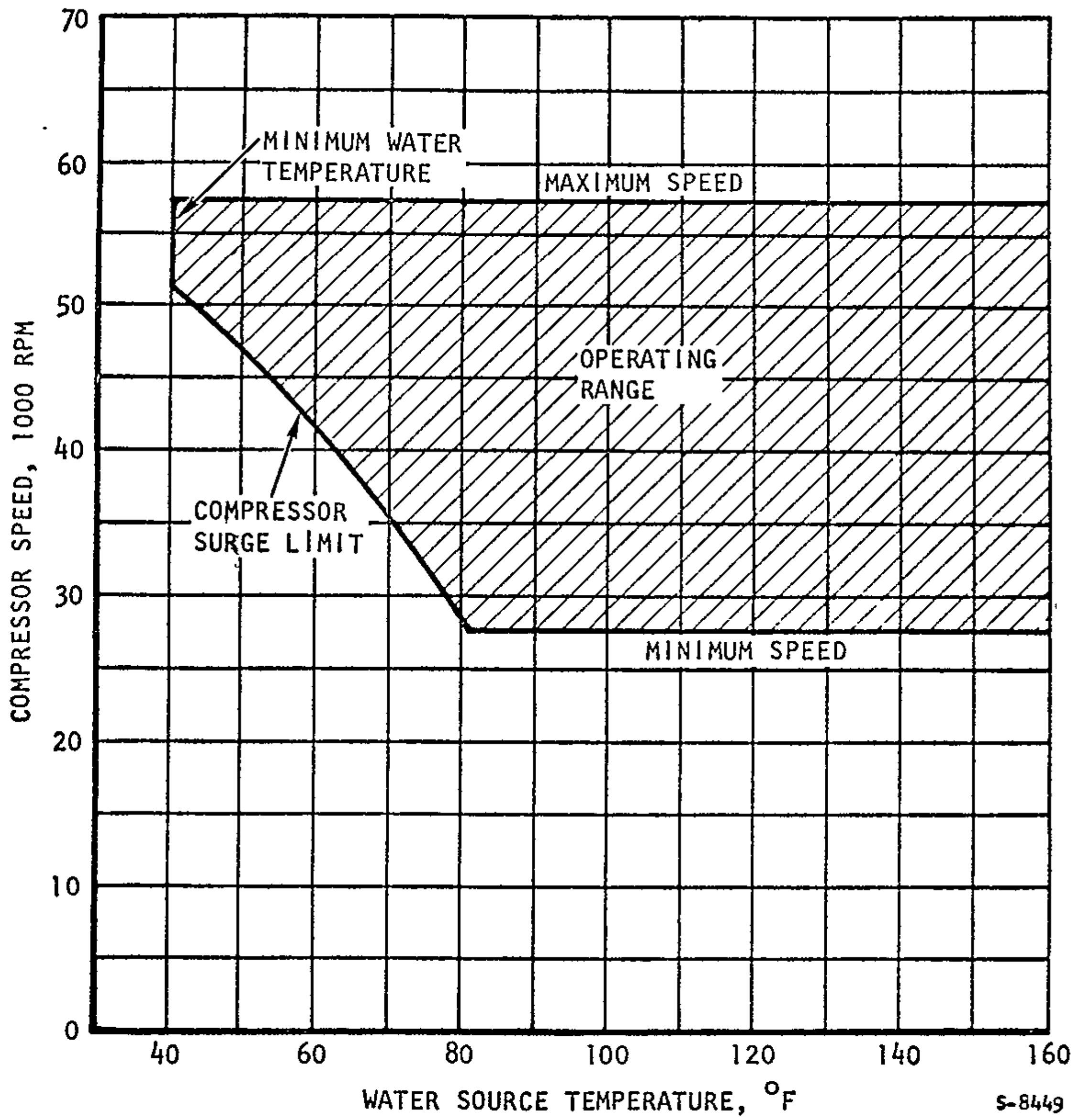

Figure 5-11. 200 KBTUH Heat Pump Operating Range

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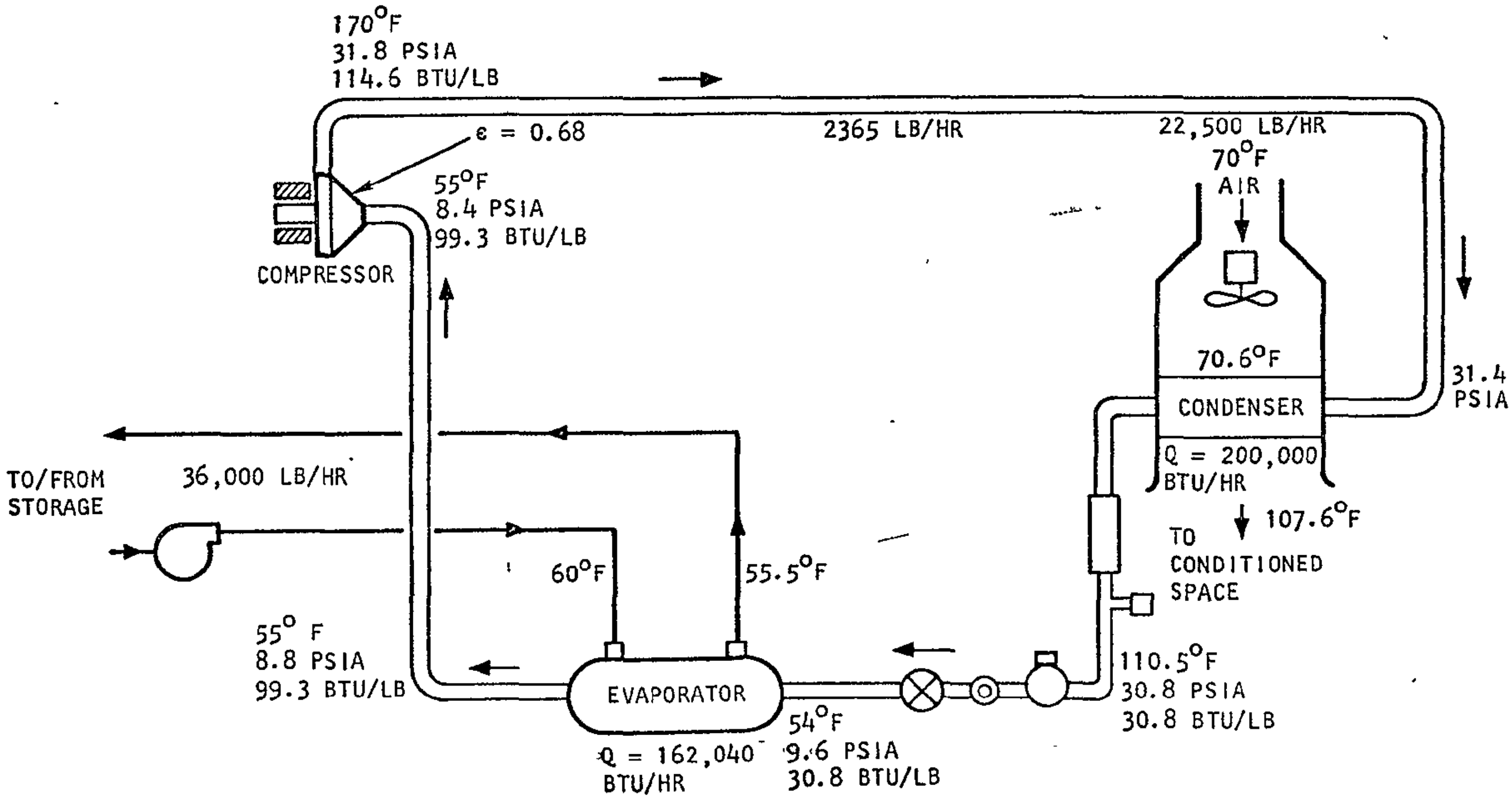

Figure 5-12. 200 KBTUH Heating Subsystem--Design Point Performance

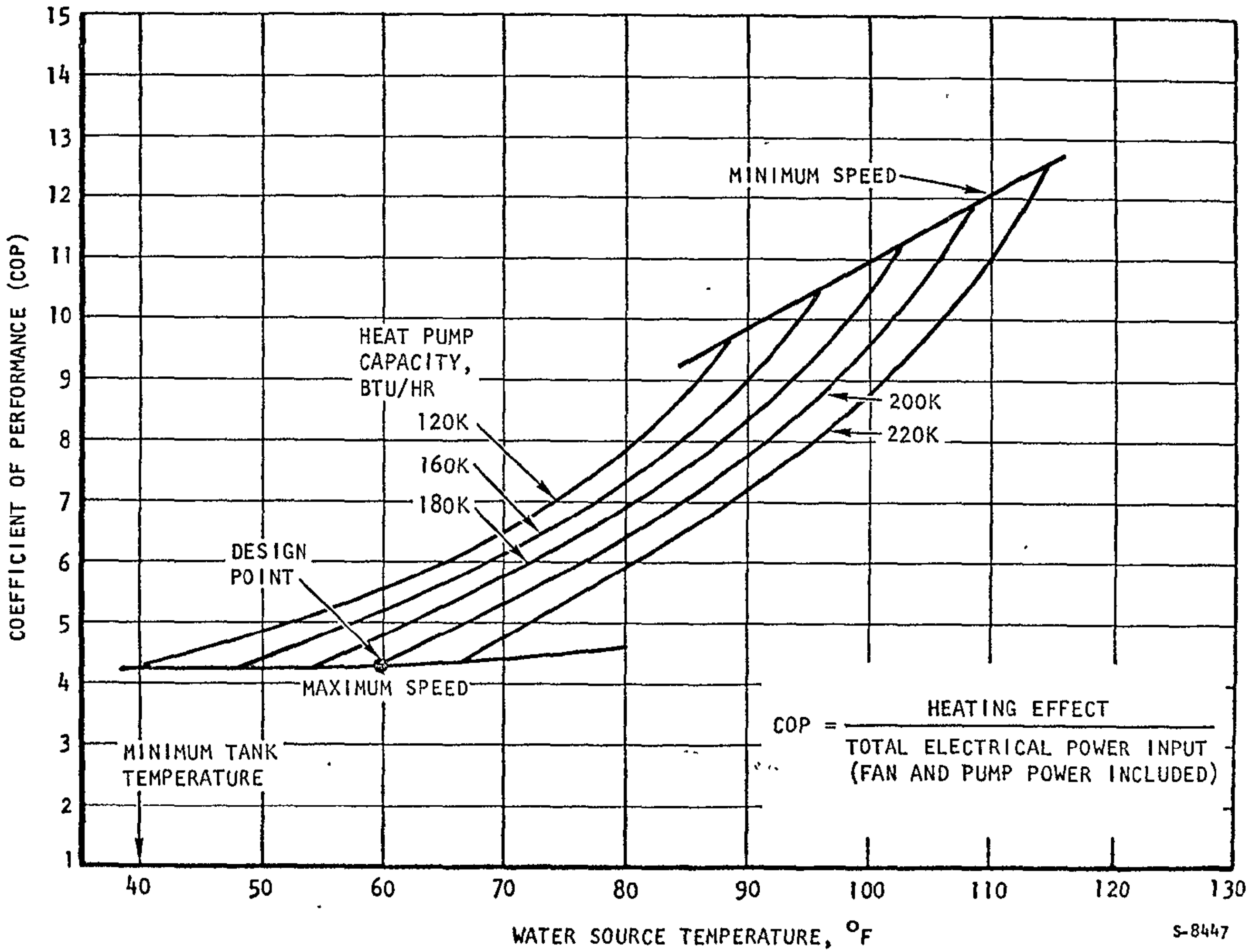

Figure 5-13. 200 KBTUH Heat Pump Coefficient of Performance

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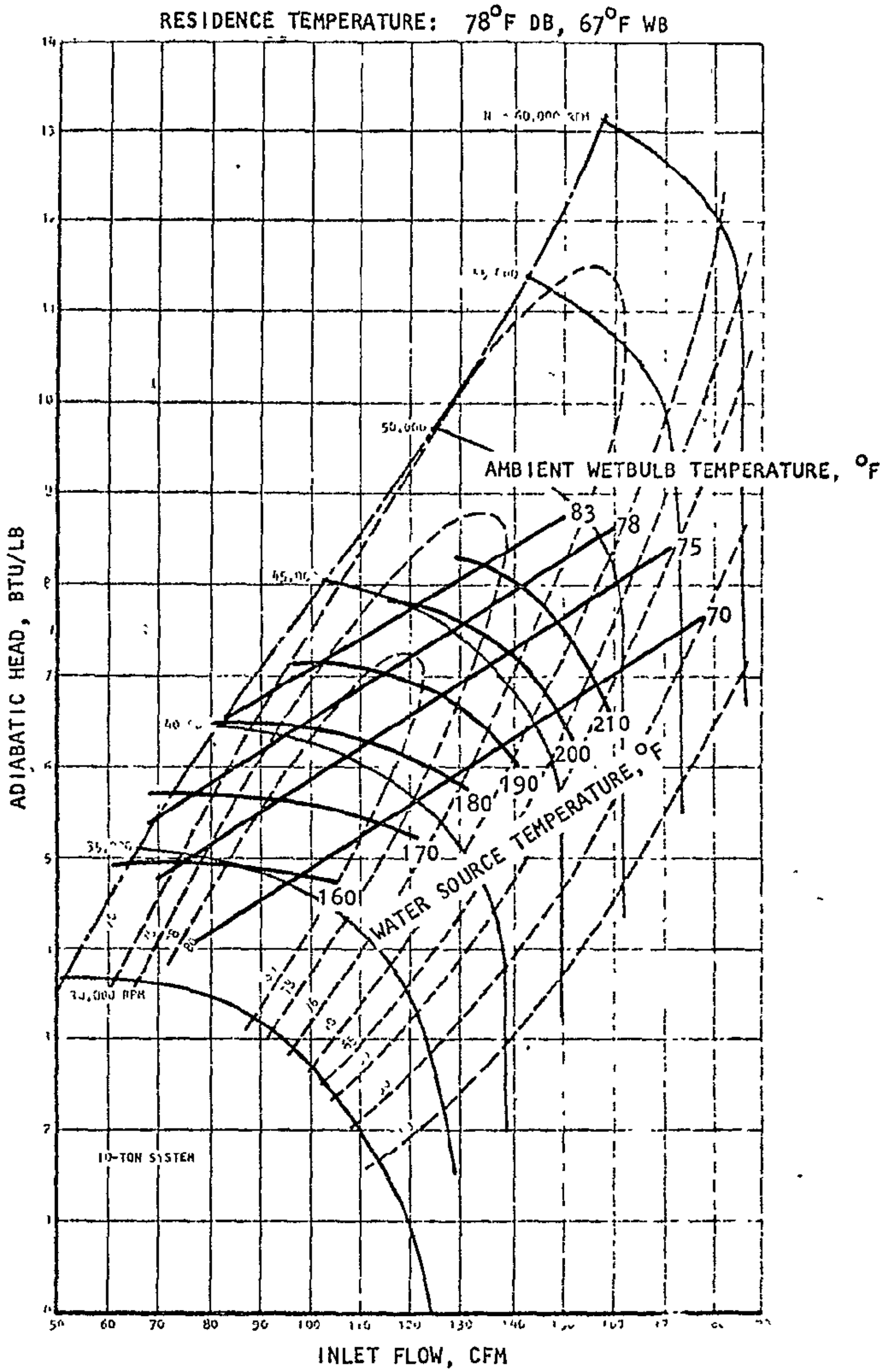

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Figure 5-14. 10-Ton Heat Pump Compressor in Cooling Mode

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5.3.2.2 Subsystem Performance

The characteristics of the heat pump for cooling mode design point are presented in Figure 5-15. These data were used for equipment design.

5.3.3 Heat Pump Package

The layout of this 800-KBTUH unit is attached to this document. Overall dimensions are 100 in. x 72 in. x 50 in. The package was designed for the heating/cooling subsystem with equipment removed for the heating-only subsystem. Again, the solar collector loop equipment (interchanger and pumps) is included. In this case the auxiliary boiler will constitute a separate package (see Section 6).

5.4 600 KBTUH/25-TON HEAT PUMP

5.4.1 Heating Mode Operation

5.4.1.1 Heat Pump Description and Control

Figure 5-16 is a schematic of this subsystem. Basically, the heat pump cycle is the same as that described above. The major difference is in the use of a water heat transport loop that collects the heat processed by the heat pump at the condenser and transports it to the terminal unit heaters. In the absence of particular sites, the number of terminal units was taken as 12.

In this case, the system is tuned to provide a 70°F average residence temperature at all terminal units. Provisions are made for adjustment of the set point temperature on the heat pump itself.

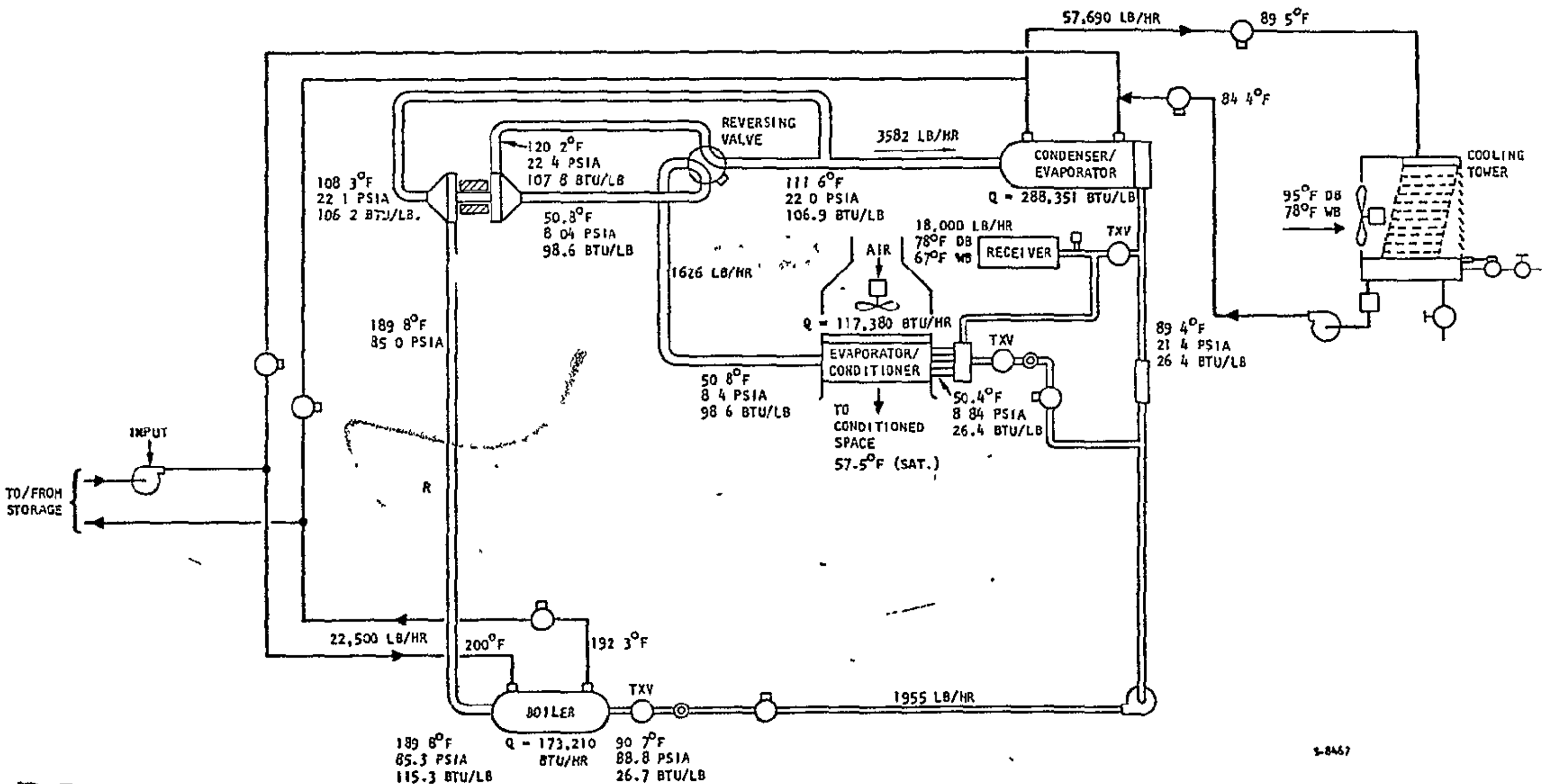

Figure 5-15. 10-Ton/200 KBTUH Cooling and Heating Subsystem--
Design Point Performance

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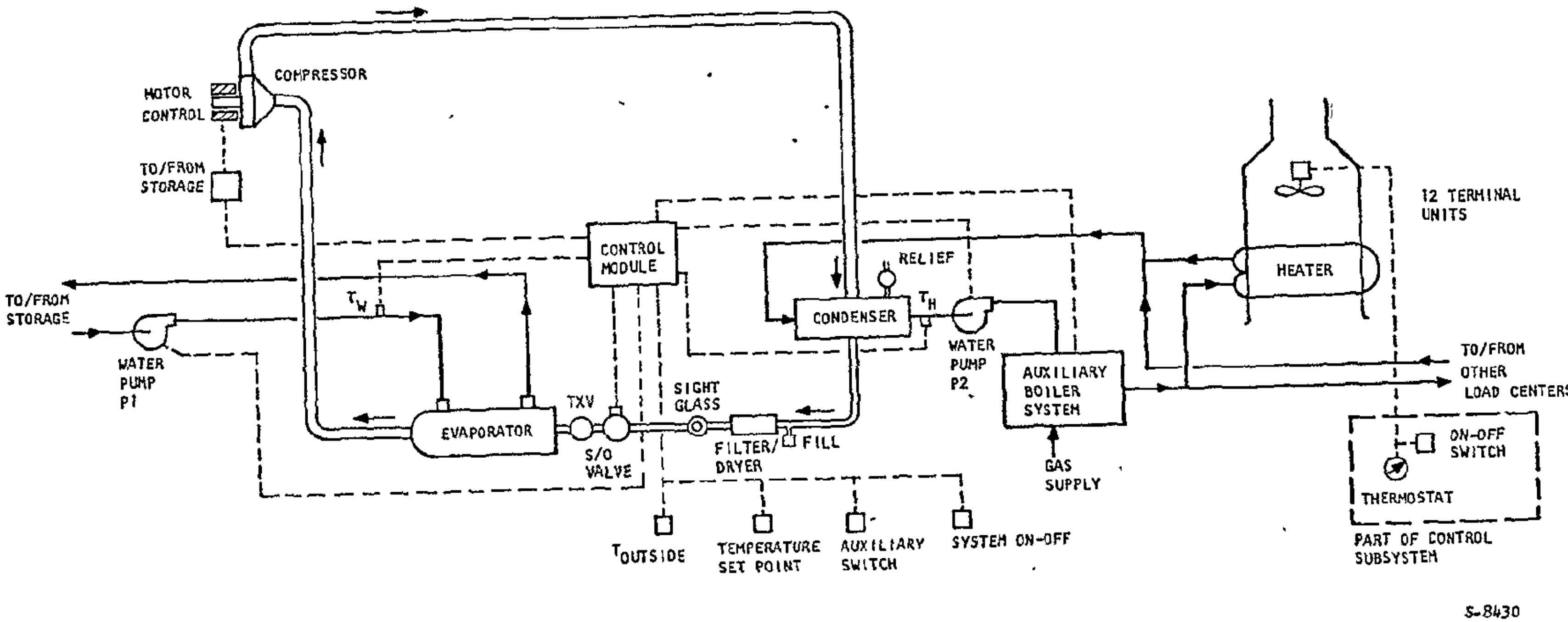


Figure 5-16. Multifamily Residence Space Heating Subsystem

The temperature of each individual load center (ILC) is controlled by a thermostat that switches the ventilation fan on and off for the center. An ON-OFF switch also is provided at each load center. The water circulation pump (P2) that ensures flow from the heat pump condenser to the terminal units is on at all times when the heating system is on.

The control concept for this heat pump is similar to that of the 60,000-Btu/hr heat pump described previously. The heat pump is operated at minimum capacity to match the heating requirements of all ILC's. In this case the water temperature at condenser outlet is used to control cycling of the heat pump on and off.

An additional sensor is used in this case to provide a reference for the control system. Outside air temperature is monitored. This parameter with the heat pump set point provides a measure of the residence heat load. This load is expressed in the form $K(T_S - T_O)$ where

K is the residence conductance, Btu/hr °F

T_S is the set point temperature

T_O is the outside air temperature

Matching the residence load with the heat pump capacity entails control of the condensing temperature of the heat pump (as water temperature at condenser outlet, T_H) so that the heating capacity is available at the terminal units. Figure 5-17 shows the required water temperature T_H as a function of heat pump capacity or residence load.

The control system uses this information in the same manner as the residence temperature in the 60 KBTUH heat pump, where T_H is the desired water temperature. The major difference is that T_H is determined from T_S and T_O. The control band

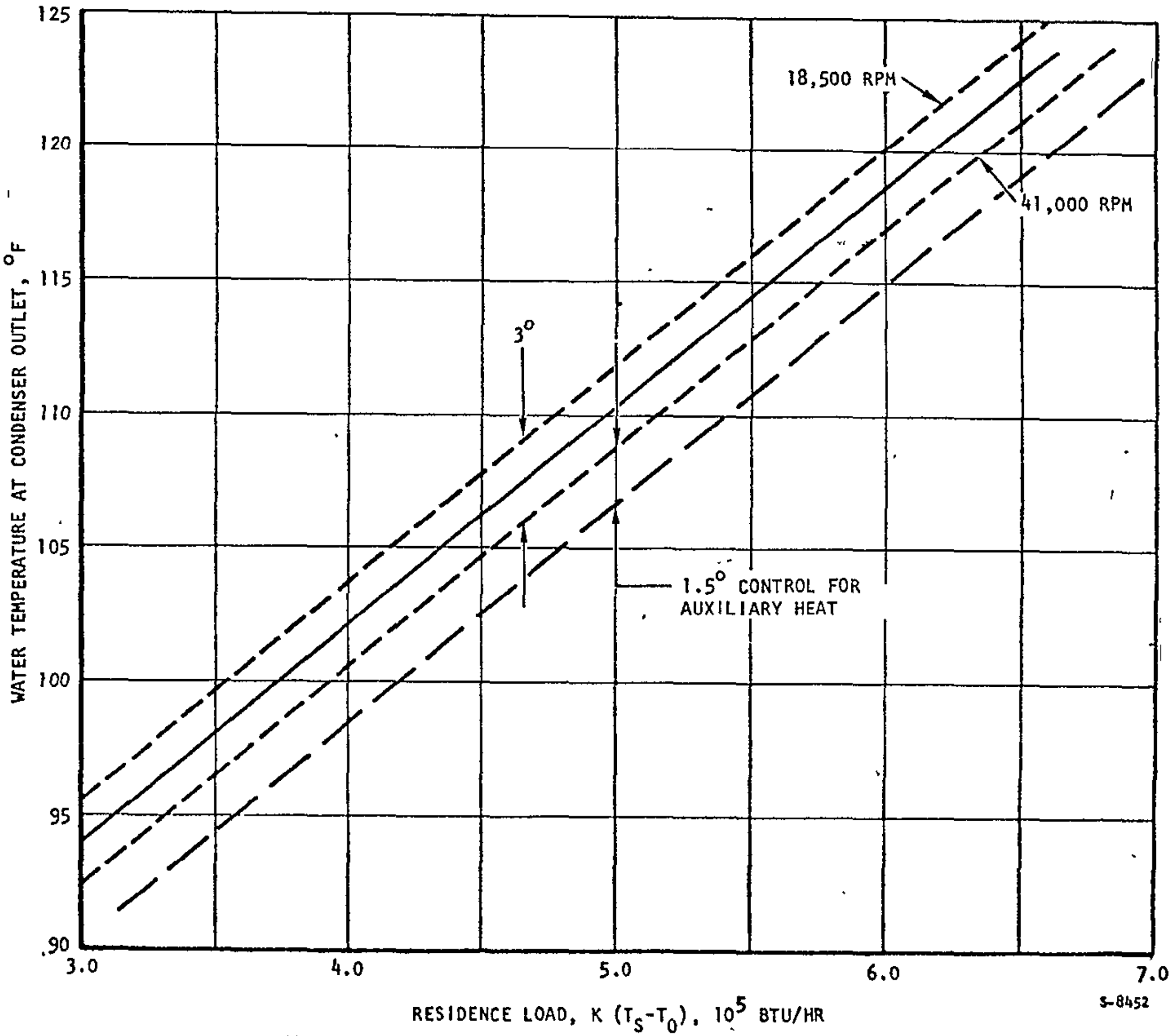


Figure 5-17. 600 KBTUH Heat Pump Control Parameters

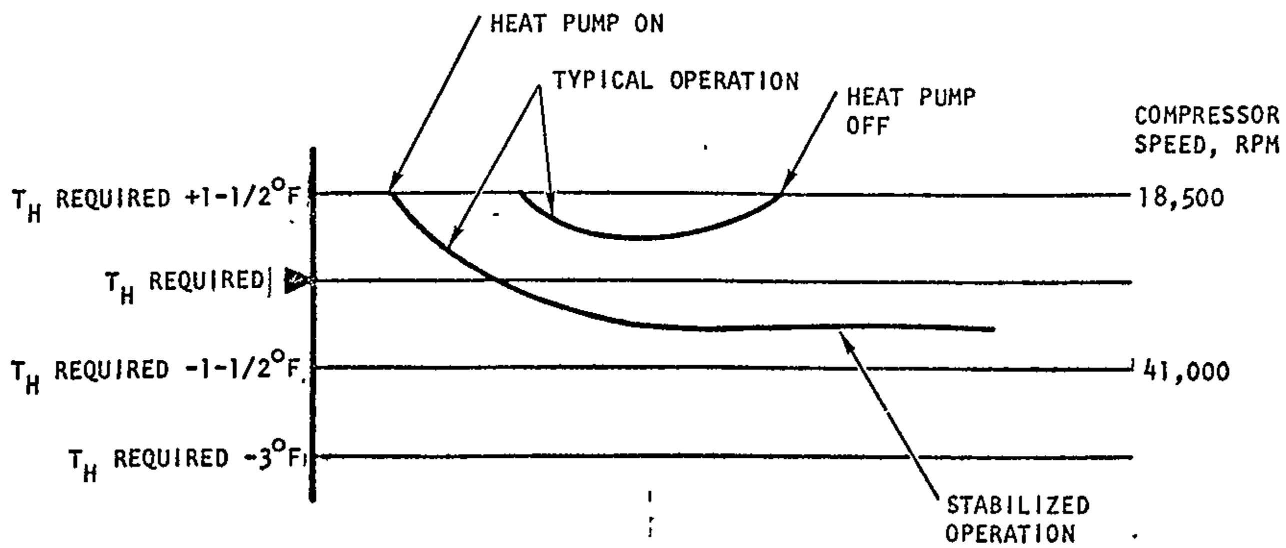
on either side of the desired T_H is $\pm$1-1/2°F. Figure 5-18 illustrates control of the heat pump in the normal mode of operation and also in the extreme case when the maximum capacity of the heat pump is exceeded. Then the auxiliary heater has to be turned on.

The operational range of the heat pump is limited by the characteristics of the compressor as discussed previously. The range of operation of the heat pump is depicted in the plot of Figure 5-19. The minimum capacity of the heat pump as a function of water source temperature can be obtained from Figure 5-19. This limitation imposed by the surge characteristics of the compressor, together with maximum and minimum operating speeds, delineates the overall operating range of the heat pump (see Figure 5-20).

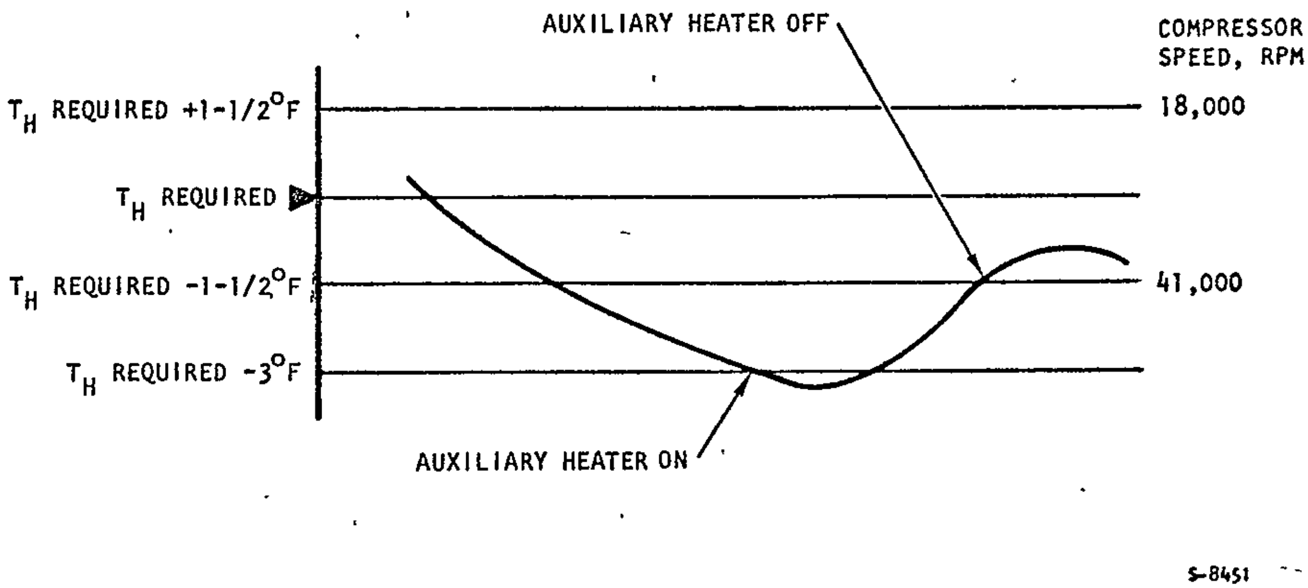
5.4.1.2 600 KBTUH Heat Pump Performance

The heat pump is designed to supply 600,000 Btu/hr at standard ARI conditions. Compressor speed under these conditions is 41,000 rpm. Figure 5-21 shows the thermodynamic characteristics of the subsystem at design point. Here again the compressor and heat exchanger performance was adjusted to the requirements of the cooling mode of operation to assure adequate operating range and maximum performance in both cases. Heat pump data in the cooling mode are presented later.

The coefficient of performance of the 600 KBTUH heat pump is shown in Figure 5-22 over the range of capacity and water source temperatures. The heat pump power input used in calculation of the COP includes the compressor motor, circulation pump motor, and water source pump motor.



a. NORMAL OPERATION



b. MAXIMUM LOAD OPERATION (LOAD > CAPACITY)

Figure 5-18. 600 KBTUH Heat Pump Control

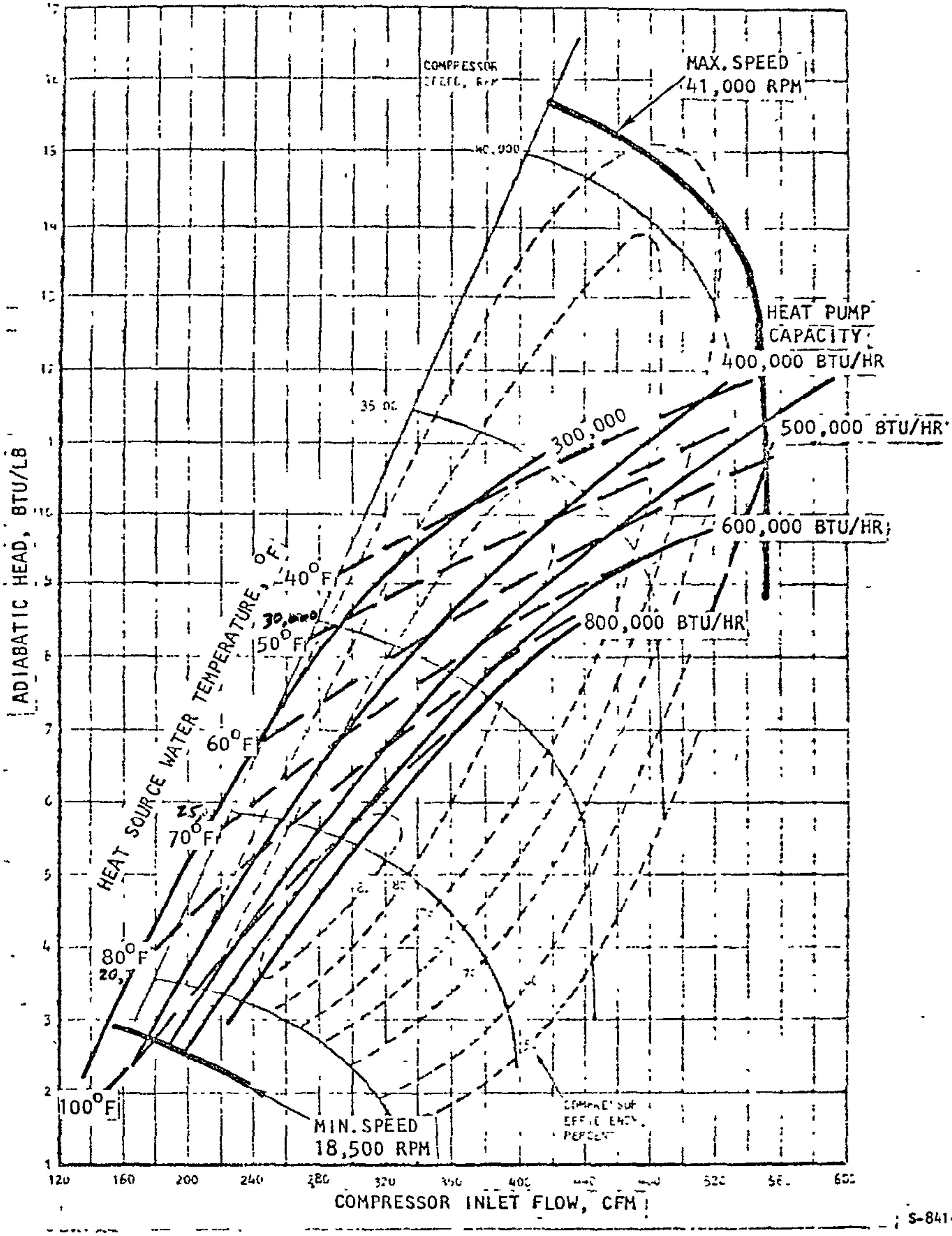


Figure 5-19. 600 KBTUH Heat Pump Compressor in Heating Mode

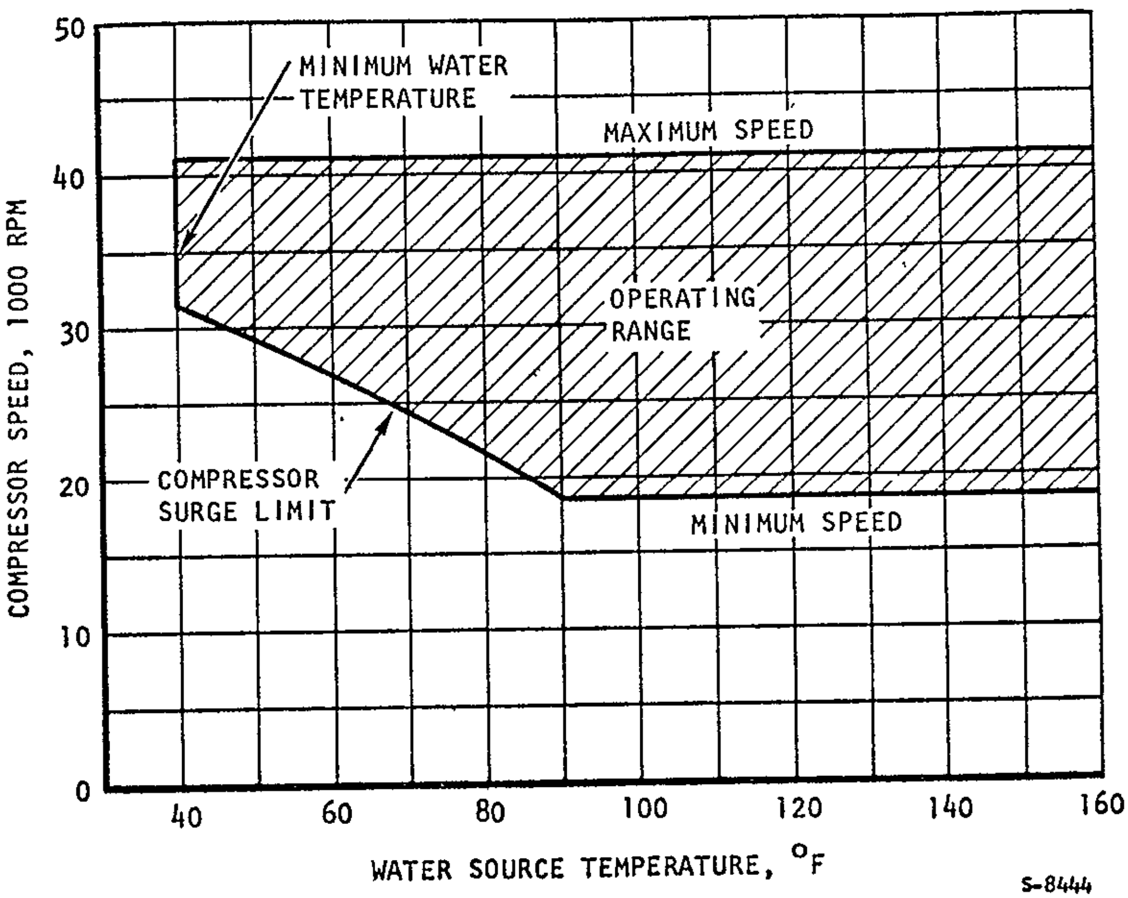


Figure 5-20. 600 KBTUH Heat Pump Operating Range

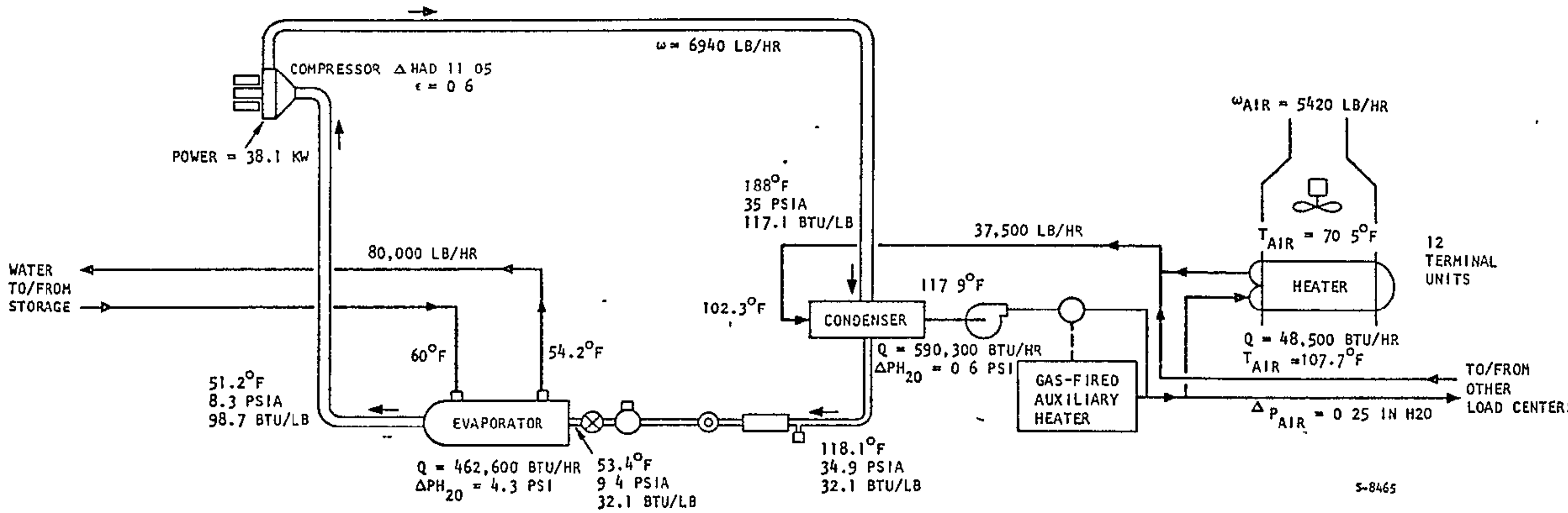


Figure 5-21. 600 KBTUH Heating Subsystem--Design Point Performance

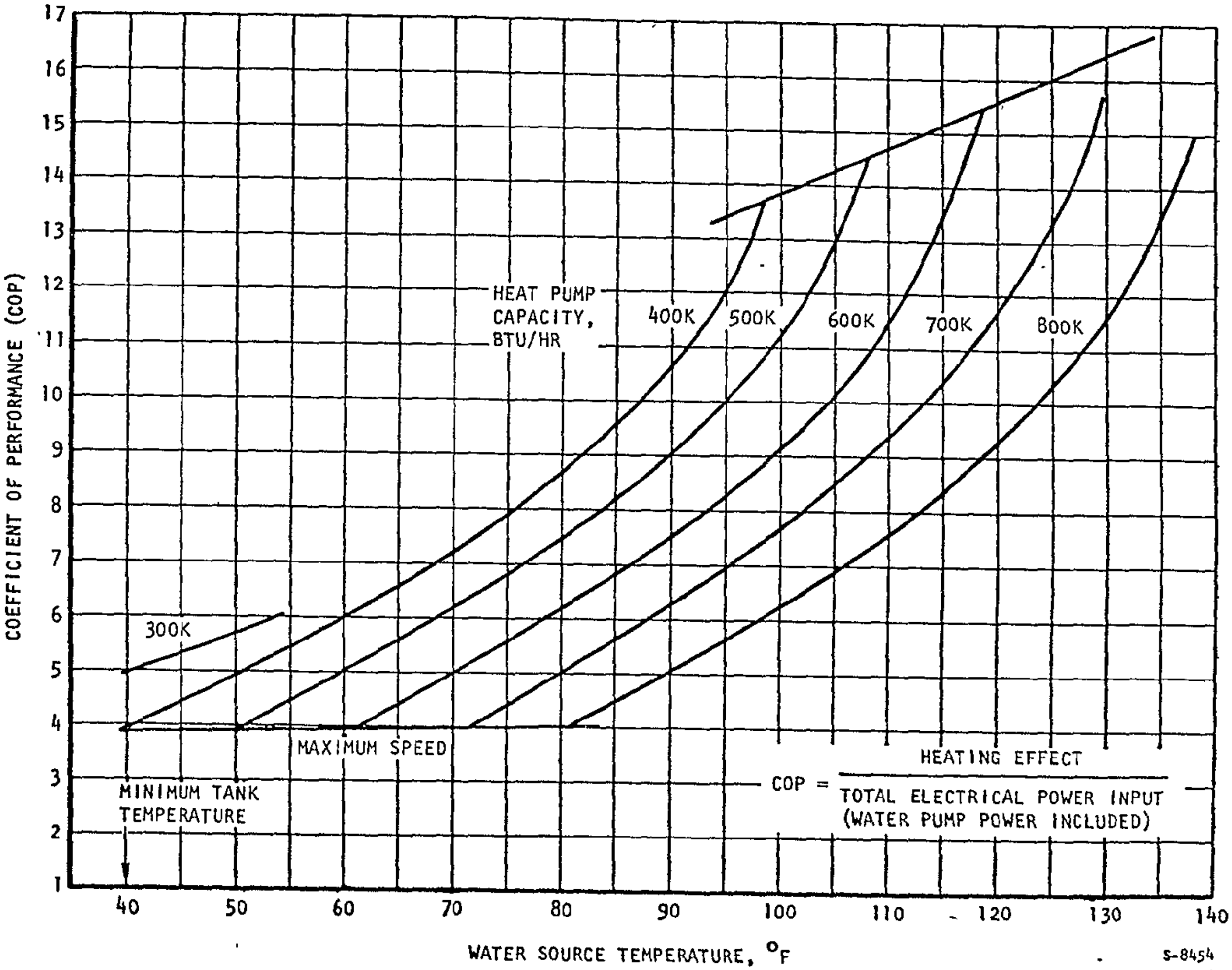


Figure 5-22. 600 KBTUH Heat Pump Coefficient of Performance

5.4.2 Cooling Mode Operation

5.4.2.1 Heat Pump Description and Control

A schematic of the heat pump with heating and cooling capability is shown in Figure 5-23. Compressor operating lines are shown in Figure 5-24.

The use of the water circulation loop to carry the heat pump effect (heating or cooling) to the 12 terminal units permits elimination of the R-11 switchover valve. In this case, water switchover valves are used so that R-11 evaporation always occurs in the same heat exchanger, as does condensation. This represents a major advantage in view of (1) the size of the switchover valve that would be required, and (2) the detail design of the heat exchangers.

Here again, a cooling tower is necessary as a heat sink in the cooling mode of operation. The tower is isolated in the heating mode and should be drained if freezing conditions are anticipated.

Seven water mode selector valves are incorporated in the design for switchover from the cooling to the heating mode. A mode selector switch is provided for this purpose. Alternatively, switchover could be made automatic using outside temperature information available to the control module.

In the heating mode, the Rankine power loop is inactive and the centrifugal compressor is motor-driven; operation and control of the subsystem in this mode have been described previously. In the cooling mode, the Rankine power loop is activated and operation is similar to that of the 3-ton air conditioner also described previously. The major difference is the use of the circulation pump, P2, between the evaporator and the twelve terminal units. A temperature sensor on the line at the evaporator outlet is used to switch the heat pump on and off. The water circulation pump, P2, is on at all times when the air conditioner is on.

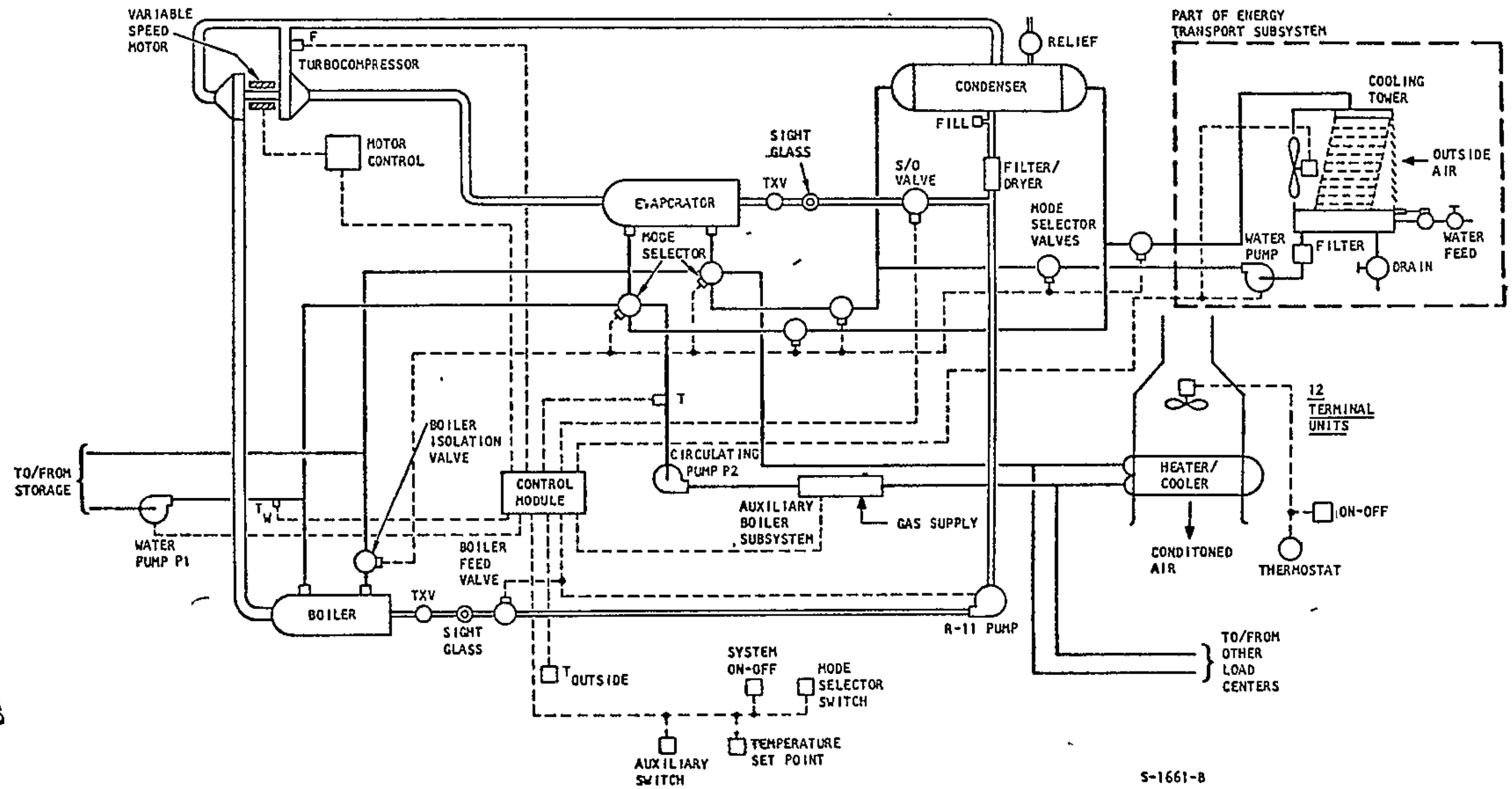


Figure 5-23. 25-Ton/600 KBTUH Cooling and Heating Subsystem

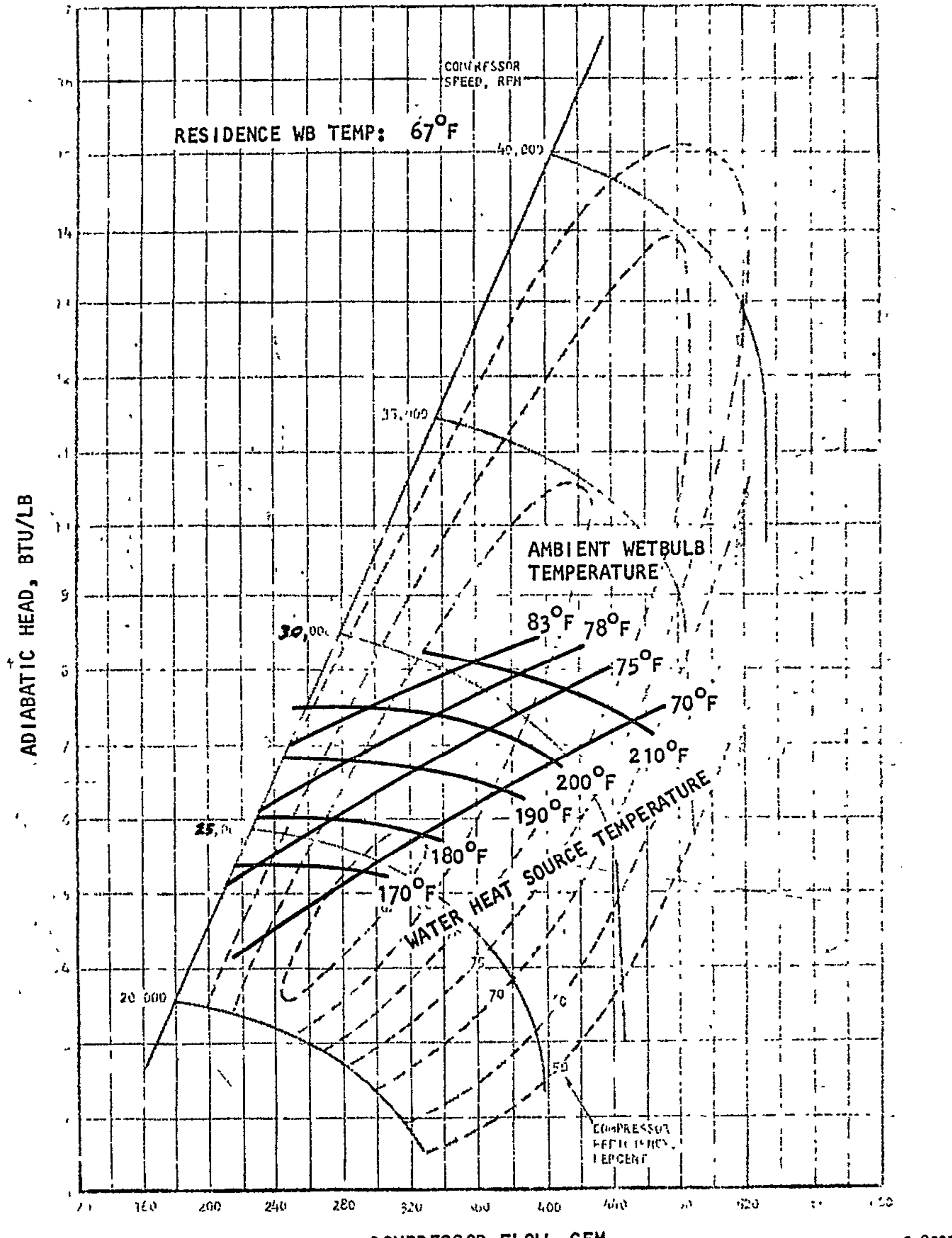


Figure 5-24. 25-Ton Heat Pump Compressor In Cooling Mode

The mode selector switch operates in the same manner as for the 3-ton/80,000-Btu/hr unit. It positions the mode selector valves and the boiler isloation valve, and interrupts power to the R-11 pump and the cooling tower.

The heating mode operation is as described previously for the 800,000-Btu/hr heating subsystem. In the air conditioning mode, subsystem on-off switching is effected by a signal from the sensors that monitor water temperature in the evaporator shell. When this temperature drops below 46°F, the system is shut off; when the water temperature increases to 50°F, the subsystem is started. Except for the start-stop signal, operation is identical with that for the 3-ton subsystem.

5.4.2.2 Subsystem Performance

The thermodynamic characteristics of the subsystem at design point are given in Figure 5-25. These data together with heating mode performance data were used in the preparation of equipment problem statements.

5.4.3 Heat Pump Package

A layout of the 200,000-BTUH heat pump package is attached to this report. This package was developed as a rooftop unit and could be installed either on the roof, on a slab adjacent to the building, or in a basement if adequate access is provided.

The package is similar in concept to that of the 100-KBTUH unit. The furnace is essentially an off-the shelf unit modified for integration into this package. Again the package was designed for heating/cooling; only the heating system components are shown. The placement of these components is similar to that of the 100 KBTUH unit.

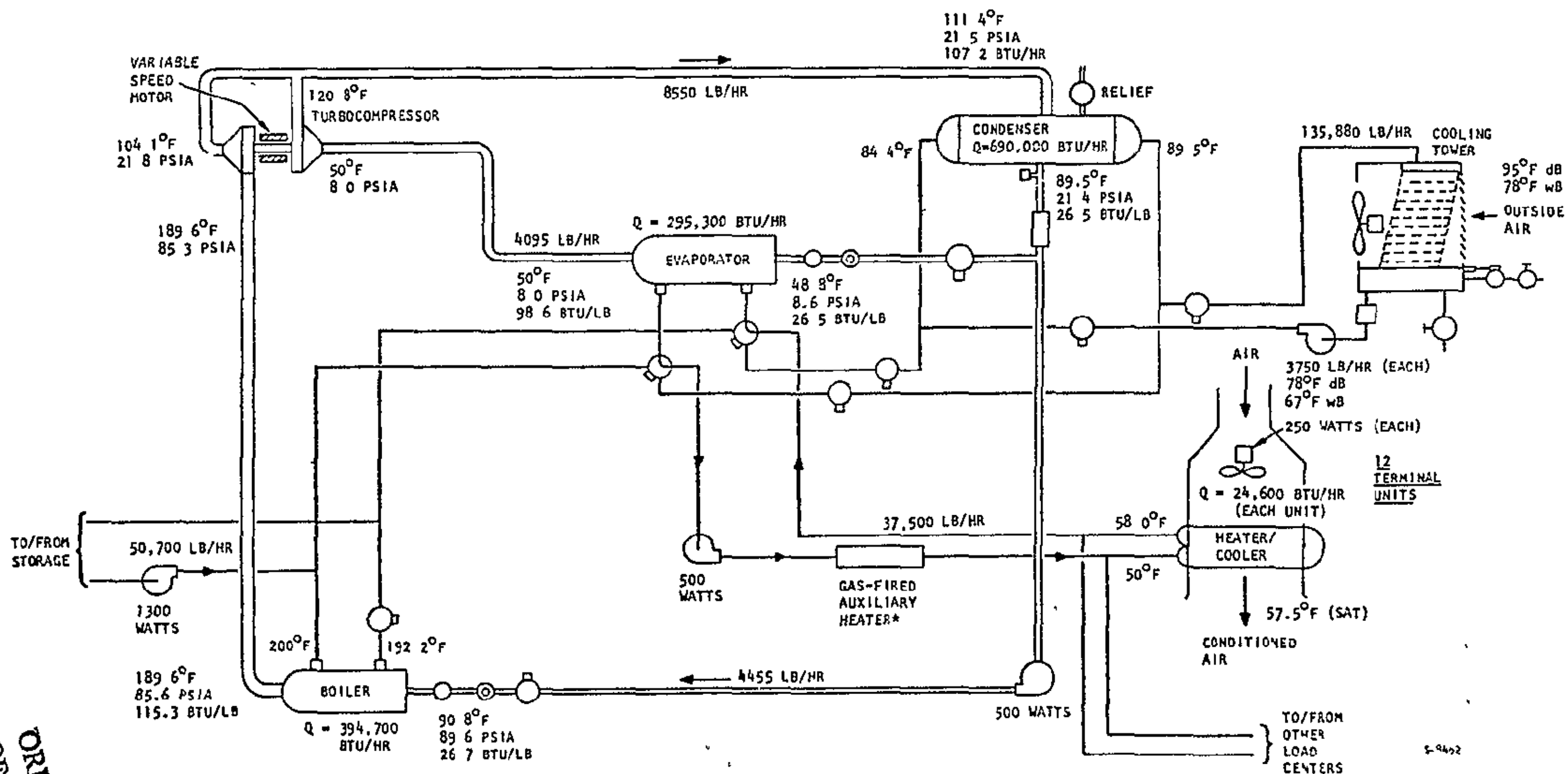


Figure 5-25. 25-Ton Cooling Subsystem--Design Point Performance

6. AUXILIARY ENERGY SUBSYSTEM

6.1 GENERAL

Solar heating systems must incorporate a means of supplementing the solar energy source when necessary because of unfavorable weather conditions and/or abnormal load situations. System-level cost trade studies have determined that the solar energy system itself should not be designed to handle the maximum load. Economic considerations dictate that the auxiliary energy subsystem capacity be considerably larger than that of the heat pump.

The single-family residential and commercial applications will be designed as rooftop units. For both sizes, gas-fired furnaces will be used as the auxiliary heating subsystems. The furnaces will be packaged as an integral part of the heat pump package as is customary for that type of system. Any size furnace could be used with the 60-KBTUH and the 200-KBUTH heat pumps. For the multifamily residence heating system, a boiler is used in the recirculation load as the auxiliary heater.

In all cases, the auxiliary energy subsystems are off-the-shelf commercially available equipment incorporating all controls and safety features required by code. Auxiliary heater activation will be from the heat pump control module as discussed in Section 5.

6.2 AUXILIARY HEATERS FOR SINGLE-FAMILY RESIDENCE AND COMMERCIAL APPLICATIONS

For the single-family residence, horizontal furnaces in the capacity range (output) of 60,000 to 150,000 Btu/hr could be employed to satisfy overall system requirements. For the commercial application, the range is about 200,000 to 300,000 Btu/hr.

The furnace will be installed downstream of the condenser coil, as depicted in Figure 6-1. The airflow will circulate through the furnace whenever the heat pump is on. The furnace is controlled on and off from the heat pump control module. No capacity modulation is used. With the furnace on, the furnace and heat pump capacities are additive. Specifications for furnaces in the range defined above are included in Exhibit 6A.

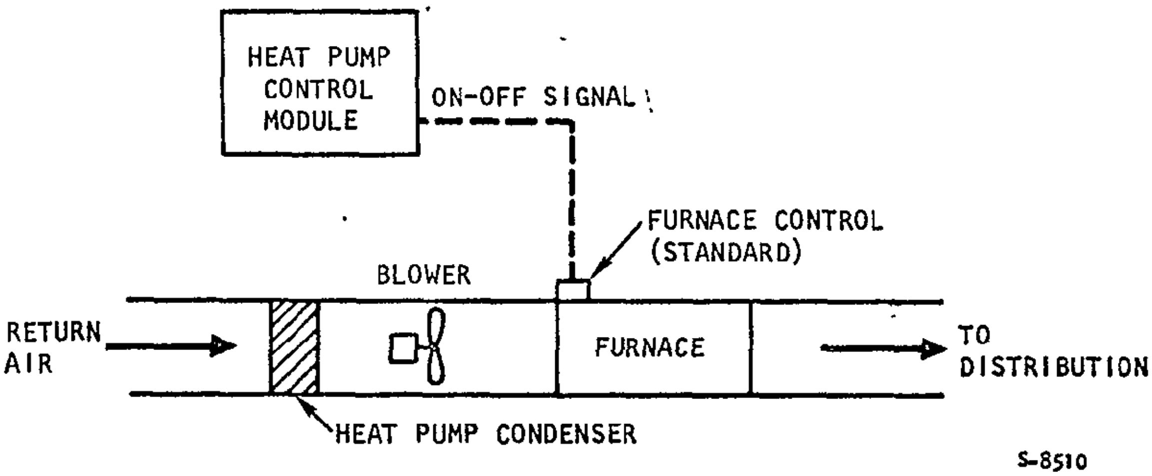


Figure 6-1. Furnace Installation (Single-Family Residence and Commercial Application)

6.3 AUXILIARY HEATER FOR MULTIFAMILY RESIDENCE

In this case, a water boiler is used consistent with the water recirculation loop. The auxiliary boiler capacity could vary over a wide range depending on the particular application. Specifications for a series of

boilers are presented at the end of this section. Boiler installation is depicted in Figure 6-2.

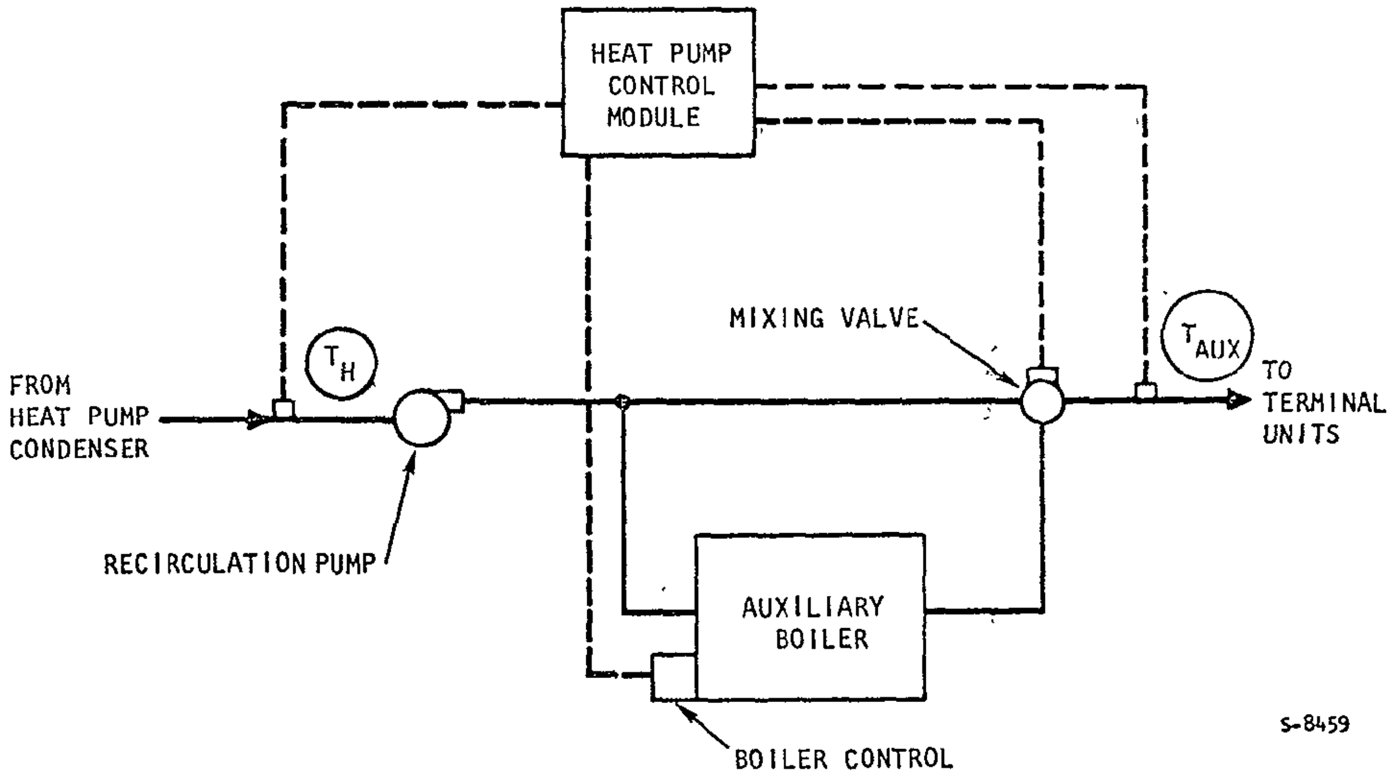


Figure 6-2. Auxiliary Boiler Installation (Multifamily Residence)

Boiler water capacity is listed in the specifications for each boiler size. The controls internal to the boiler are standard equipment; it is not intended to change any of these controls involving safety of operation and water temperature control. A turndown ratio of 2:1 will be provided to reduce boiler on-off cycling to a minimum.

The boiler will be turned on and off using a signal from the heat pump control module. The information used by the control module to provide this signal is the water temperature at heat pump condenser outlet (T_H). Should

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this temperature drop to a value 3°F below the T_H set point, the boiler will be activated at minimum firing rate. The heat pump control module incorporates circuitry to operate the mixing valve so that the following condition is maintained:

$$T_H - 3°F < T_{AUX} < T_H - 1\ 1/2°F$$

Should the boiler output exceed the demand, the water temperature within the boiler will increase to a maximum value of 200°F. At this temperature, the boiler control will override the heat pump control module signal and shut down the boiler. The boiler will remain off until the boiler water temperature drops to 170°F. At this point, the boiler will be fired again if commanded by the heat pump control.

Boiler capacity will increase as the water temperature in the boiler drops below 170°F.

EXHIBIT 6A

DUNHAM-BUSH BOILERS

IRON FIREMAN

Packaged Boilers

Model 36-45 Series

15 PSIG Steam or 30 PSIG Water
#2 & #4 Oil, Gas or Gas/Oil

13 4 thru 150 8 Horsepower

COMPARE THESE DESIGN FEATURES·

- 80% Plus Combustion Efficiency
- Low Electrical Requirements
- Compactness of Package
- No Refractory Baffles
- Easy Access For Maintenance
- Only A Stub Stack Needed

Features

The 36 45 Series 3 pass packaged boiler is designed to produce 80% plus firing efficiency Overall economy is assured by sealed tight combustion equipment with very low off period standby loss

Only a vent or stub stack is necessary since the boiler is "pressurized" so that no flue gases can leak into the room - all joints are welded or gasketed

The large furnace provides sufficient volume for efficient combustion and a large amount of primary heating surface Moderate firing provides reliable operation with minimum maintenance

Electrical horsepower and combustion blower noise level are minimized by the generous construction and consequent low draft loss through the boiler

Compact design permits installations in restricted areas Low water level eliminates necessity for high ceilings

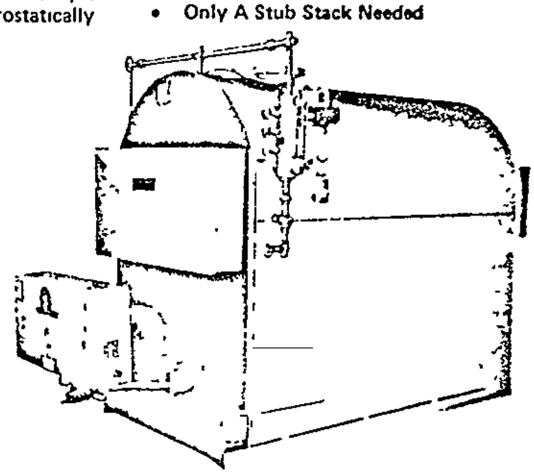
Installation is simplified by the standard complete packaging The boiler can be supplied in assemblies to fit any field installation conditions

The ASME Seal of the American Society of Mechanical Engineers on the boiler and the Underwriters' Laboratories, Inc label on the burner are assurance of the highest standards of quality in design, materials and workmanship

Each Boiler is designed for 15 psi steam or 30 psi water, hydrostatically tested at 60 psi.

APPROVALS

Underwriters Laboratories Inc.
LISTED



Ratings & Capacities

PACKAGED BOILER MODEL NO 36-45		63	78	92	107	127	154	180	209	239	275	300	343	OHIO Special*	450	562	675
Certified Gross Rating	Horsepower	13 4	16 4	19 4	22 4	28 4	34 4	40 3	46 3	52 3	61 2	70 0	80 0	100 0	100 0	126 9	150 8
Steam (from & at 212°F)	Lbs/Hr	464	567	670	773	979	1186	1392	1598	1804	2113	2423	2737	3452	3452	4380	5204
Steam or Water	MBh	450	550	650	750	950	1150	1350	1550	1750	2050	2350	2650	3350	3350	4250	5050
Gross Rating Steam (240 Btu)	Sq Ft	1875	2291	2708	3125	3958	4792	5625	6458	7292	8542	9791	11042	13958	13958	17708	21042
Firing Rate Gas	MBh	563	688	813	938	1188	1438	1688	1938	2188	2563	2937	3313	4181	4181	5312	6312
No 2 Oil (140 000 Btu/Gal)	Gph	4 0	4 9	5 8	6 7	8 5	10 3	12 1	13 8	15 6	18 3	21 0	23 6	30 0	30 0	38 0	45 0
No 4 Oil (144 000 Btu/Gal)	Gph	3 9	4 8	5 6	6 5	8 3	10 0	11 8	13 4	15 2	17 8	20 4	23 0	29 2	29 2	36 9	43 8
Heating Surface Fireside	Sq Ft	59	72	85	98	118	142	166	192	220	253	284	315	315	418	521	619
Waterside	Sq Ft	63	78	92	107	127	154	180	209	239	275	309	343	343	450	562	675
Total Furnace Volume	Cu Ft	7 8	9 1	10 5	11 7	15 4	17 9	20 4	23 3	29 5	33 4	37 1	40 8	40 8	66 0	60 4	94 3
Heat Release Rate	MBh/Cu Ft/Hr	72 2	75 6	77 4	80 2	77 1	80 3	82 7	83 2	74 2	76 7	79 2	81 2	102 5	63 3	66 1	66 9
Safety Valve Capacity Min	Lbs/Hr	504	624	736	856	1016	1232	1440	1672	1910	2200	2472	2744	3600	3600	4496	5400
Water Content Steam Boiler	Gal	68	83	98	112	120	145	169	196	223	256	288	319	319	469	582	689
Water Boiler	Gal	92	111	132	151	157	188	220	254	286	328	368	408	408	620	768	906
Approx Weight Filled to NWL Steam	Lbs	2450	2750	3000	3300	3600	4150	4700	5300	5850	6600	7300	7950	7950	12100	14150	16550
Approx Weight Filled Hot Water	Lbs	2650	3000	3300	3600	3900	4500	5150	5750	6400	7200	8000	8700	8700	13400	15700	18400
Approx Shipping Weight	Lbs	1900	2050	2200	2350	2600	2950	3300	3650	4000	4450	4900	5300	5300	8200	9300	10800

* OHIO SPECIAL MODEL NO IS 36 34 343 (FOR STATE OF OHIO ONLY)

DUNHAM-BUSH, INC. • Harrisonburg, Virginia 22801, U.S.A.

one of The Signal Companies [S]

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Standard Equipment - - All Models

3 Pass forced draft boiler
ASME constructed firebox boiler
Burner w/manual On Off switch
Seal welded skid type base
Refractory lined firebox floor
Horizontal smoke outlet w/cleanout
Pressure tight front flue door
Flame observation port, pyrex
Jacket with insulation
Painted finish
Supply and Return connections
Flue cleaning brush and rod (oil only)
Washout tappings w/plugs.
Washout plug wrench

Signal Lights For ignition, main fuel(s) and Lockout

STEAM TRIM inc
- M&M #767 LWCO* (36-45-63 thru 36 45 343 boilers)
- Magnetrol # W-163 LWCO (36 45-450 and larger boilers)
- ASME safety valve(s)
- Compound steam gauge
- Water gauge glass
- Drain Valve on Water Column
- Limit control
- Operating control

*The low water cut-off is for boiler protection only and can not be used as a feeder

2" boiler tubes
Lifting Lugs on 36 45 450 and larger boilers
Access plate to firebox area in 36 45 450 and larger boilers

WATER TRIM inc
- M&M #764 LWCO* (36 45 63 thru 36 45 343 boilers)
- M&M #64 LWCO* (36 45-450 and larger boilers)
- ASME relief valve (s)
- Press Temp gauge
- Drain Valve on LWCO
- Limit Aquastat
- Operating aquastat

Standard Equipment - - #2 Oil — Whirl Power Models

PACKAGED BOILER MODEL NUMBER FORCED DRAFT PRESSURE ATOMIZING	36-45 63 36-45-78 36-45 92 36-45-107	36-45 127	36-45-154 36-45-180	36-45-209	36-45 239	36-45-275	36-45-309 36-45-343 36-45-450 36-34 343	36-45-562
Burner Model No Used	MF 65E-A	CF-100	C-120 1 8	C-120-2 1	C 120-C 2 5	C 240 SM E	C 240-SM F	C 240-SM K
Burner Catalog No Used	149162-G01	149144-GO1	144989 GO3	144989 G52	144995-G52	149117 GO1	149117 GO5	149077 GO1
Fire Control	LFS	LFS	LFS	LFS	LFS	LFS	LFS	LFS
Combustion Safeguard	TFC-2	UVC2	UVC2	UVC2	UVC2	UVC2	UVC2	5023
Blower Motor Horsepower	1/4	1/2	3/4	1	1	1 1/2	2	3
Motor and Fan RPM	3450	3450	3450	3450	3450	3450	3450	3450
Motor Starter or Relay	Std	Std	Std	Std	Std	Std	Std	Std
Pre and Post Purge	Std	Std	Std	Std	Std	Std	Std	Std
Control Panel	Std	Std	Std	Std	Std	Std	Std	Std
Fuel Unit (Pump)	2 - Stage 300 psi	2 - Stage 300 psi	2 - Stage 300 psi	2 - Stage 300 psi	2 - Stage 300 psi	2 - Stage 300 psi	2 - Stage 300 psi	2 - Stage • 300 psi •
Ignition Transformer	Std	Std	Std	Std	Std	Std	Std	Std
Standard Electric (Motor)	115/60/1	115/60/1	115/60/1	230/60/3	230/60/3	230/60/3	230/60/3	230/60/3
Standard Electric (Controls)	115/60/1	115/60/1	115/60/1	115/60/1	115/60/1	115/60/1	115/60/1	115/60/1
Control Circuit Transformer	—	—	—	Std	Std	Std	Std	Std

• The 36-45-562 is equipped with gas-electric ignition, including pilot manual valve, pressure regulator and solenoid valve. Also 3/4 hp (230/60/1) oil pump mounts separate

Standard Equipment - - Gas — Whirl Power Models

PACKAGED BOILER MODEL NUMBER FORCED DRAFT POWER TYPE	36-45-63 36-45-78 36-45 92	36-45 107 36-45 127	36-45-154 36-45-180 36-45 209	36-45-239	36-45-275	36-45-309 36-45-343 36-45-450 36 34-343	36-45-562
Burner Model No Used	MF 85G A	CF-100-G	C-120G-2 1	C-120G C-2 5	C 240G SM E	C 240G SM F	C 240G SM K
Burner Catalog No Used	149169 GO1	149150-GO1	144999-GO3	149004-G57	149119 GO1	149119 GO9	149075 GO7
Fire Control	On-Off	On Off	On Off	LFS	LFS	LFS	Full Mod
Combustion Safeguard	TFC-2	UVC2	UVC2	UVC2	UVC2	UVC2	5023
Blower Motor Horsepower	1/4	1/2	1/2	3/4	1 1/2	2	3
Motor and Fan RPM	3450	3450	3450	3450	3450	3450	3450
Motor Starter or Relay	Std	Std	Std	Std	Std	Std	Std
Pre and Post Purge	Std	Std	Std	Std	Std	Std	Std
Control Panel	Std	Std	Std	Std	Std	Std	Std
Standard Gas Pipe Group IPS	1 1/4"	1-1/4	2"	2	2-1/2"	3	3
Standard Piping Group No	145299 GO1	145301 GO1	145346 GO2	145346 GO2	145491 GO1	145493 GO2	145493 GO1
Gas Pressure Required - W C	4 9"	7 2"	5 3"	6 7"	5 4	7 0 •	9 4
Main Automatic Gas Valves (two)	Std	Std	Std	Std	Std	Std	Std
Pilot and Main Manual Gas Cock	Std	Std	Std.	Std	Std	Std	Std
Pilot and Main Gas Pressure Reg	Std d	Std d	Std d	Std d	Std d	Std d	Std d
Pilot Solenoid Valve	Std	Std	Std	Std	Std	Std	Std
Air Proving Interlock	Std	Std	Std	Std	Std	Std	Std
Ignition Transformer	Std	Std	Std	Std	Std	Std	Std
Standard Electric (Motor)	115/60/1	115/60/1	115/60/1	115/60/1	230/60/3	230/60/3	230/60/3
Standard Electric (Controls)	115/60/1	115/60/1	115/60/1	115/60/1	115/60/1	115/60/1	115/60/1
Control Circuit Transformer	—	—	—	—	Std	Std	Std

• The 36-34 343 Ohio Special Boiler requires 9 2" W C., not 7 0" W C

d• Deleted with L P. gas units. Normally Furnished by gas supplier

— 36 Series Boilers are equipped to burn No 2 oil and/or gas. Reference to gas is based on Natural Gas (1000 Btu content: 0.60 specific gravity) For other gas fuels consult the Application Engineering Department

FOR ADDITIONAL INFORMATION CONSULT YOUR LOCAL IRON FIREMAN SALESMAN

All information contained herein is for reference only and is subject to change without notice

Standard Equipment - - Gas / #2 Oil – Whirl Power Models

PACKAGED BOILER MODEL NUMBER FORCED DRAFT PRESSURE ATOMIZING	36-45-63 36-45 78 36-45 92 36-45 107 36-45 127	36-45 154 36-45-180	36-45 209	36-45-239	36-45 275	36-45 309 36-45 343 36-45 450 36 34 343	36-45-562
Burner Model No. Used	CF-100-GO	C 120GO-1 8	C 120GO 2 1	C-120GO C 2 5	C 240GO SM I	C 240GO SM F	C 240GO SM K
Burner Catalog No Used	149154-GO1	149008-GO1	149008-G19	149013-G57	149121 GO1	149121 GO9	149132 GO7
Fire Control - - Oil	LFS	LFS	LES	LFS	LFS	LFS	LFS
- -Gas	On-Off	On Off	On Off	LFS	LFS	LFS	Full Mod
Combustion Safeguard	UVC2	UVC2	UVC2	UVC2	UVC2	UVC2	5023
Blower Motor Horsepower	1/2	3/4	3/4	1	1 1/2	2	3
Motor and Fan RPM	3450	3450	3450	3450	3450	3450	3450
Motor Starter or Relay	Std	Std	Std	Std	Std	Std	Std
Pre and Post Purge	Std	Std	Std	Std	Std	Std	Std
Control Panel	Std	Std	Std	Std	Std	Std	Std
Fuel Unit (Pump)	2 - Stage 300 psi	2 - Stage 300 psi	2 - Stage 300 psi	2 - Stage 300 psi	2 Stage 300 psi	2 Stage 300 psi	2 Stage • 300 psi •
Ignition Transformer	Std	Std	Std	Std	Std	Std	Std
Standard Gas Pipe Group I P S	1 1/4"	2"	2	2	2 1/2	3	3
Standard Piping Group No	145301 GO1	145346 GO2	145346 GO2	145346 GO2	145491 GO1	145493 GO2	145493 GO1
Gas Pressure Required W C	7 2'	4 5'	5 3	6 7'	5 4"	7 0" ••	9 4
Main Automatic Gas Valves (two)	Std	Std	Std	Std	Std	Std	Std
Pilot and Main Manual Gas Cock	Std	Std	Std	Std	Std	Std	Std
Pilot and Main Gas Pressure Reg	Std d	Std d	Std d	Std d	Std d	Std d	Std d
Pilot Solenoid Valve	Std	Std	Std	Std	Std	Std	Std
Air Proving Interlock	Std	Std	Std	Std	Std	Std	Std
Standard Electric (Motor)	115/60/1	115/60/1	115/60/1	230/60/3	230/60/3	230/60/3	230/60/3
Standard Electric (Controls)	115/60/1	115/60/1	115/60/1	115/60/1	115/60/1	115/60/1	115/60/1
Control Circuit Transformer	--	--	--	Std	Std.	Std	Std

• The 36-45 562 is also equipped with a 3/4 hp (230/60/1) oil pump that mounts separately

•• The 36-34 343 Ohio Special Boiler requires 9 2"W C , not 7 0 "W C

d= Deleted with L. P. gas units. Normally furnished by gas supplier

NOTE Gas/Oil Models are equipped with gas-electric ignition for both - gas and oil firing with C-120 & C 240 burners

Standard PAQ-PAGO-AG Burner Models used with 36-45 Boiler Series (#2 & #4 oil)

Boiler Number	BURNER MODEL NUMBER					ELECTRICAL		
	No 2 Oil	No 4 Oil	Gas/No 2 Oil	Gas/No 4 Oil	Gas	Blower Motor (H P)	Oil Pump Motor H P	Heater For No 4 Oil (kw)
36-45 309	PAO 2 4 5	PAO 4 4 5	PAGO 2 4 5	PAGO 4 4,5	N/A	3	•	4
36-45 343	PAO 2 4 5	PAO 4 4 5	PAGO 2 4,5	PAGO 4 4,5	N/A	3	•	4
36-45-450	PAO 2 4 5	PAO 4 4 5	PAGO 2 4,5	PAGO 4 4 5	N/A	3	•	4
36-45 562	PAO 2 6 3	PAO 4 6 3	PAGO 2 6,3	PAGO 4 6 3	AG 6 3	5	¾	5
36-45 675	PAO 2 6,3	PAO 4 6,3	PAGO 2 6,3	PAGO 4 6,3	AG 6,3	5	¾	5

N/A – Not Available, refer to other Iron Fireman literature for availability
•– The oil pump is driven from the same motor that drives the fan (only one motor starter is required and furnished).

Optional Equipment - - at extra cost

Alarms
Alternate Water Level Controls
Automatic Fuel Changeover
Boiler - - "ONLY"
60 p s i Working Water Pressure
Dip Tube

Domestic Hot Water Heater
Explosion Relief Door
Factory Insurance Association
Factory Mutuals

Additional Connections
Lifting Lugs*
Rear Access Door
Signal Lights (extra)
Stack Thermometer
Vertical Flue Outlet

*Boilers smaller than 36 45 450

Information Required When Ordering

1 Boiler Model Number?
2 Steam or Water?
3 Gas or Oil or Gas/Oil?
4 Electrical Characteristics?
5 Gas Type Btu/cf and Pressure Available at boiler?
6 Approximate Shipping Date Required?
7 Delivery Via?
8 Insurance Option - UL FM FIA?

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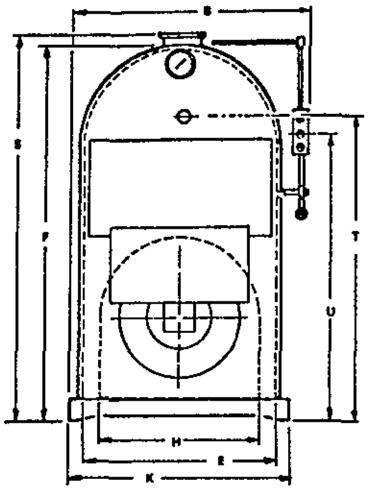

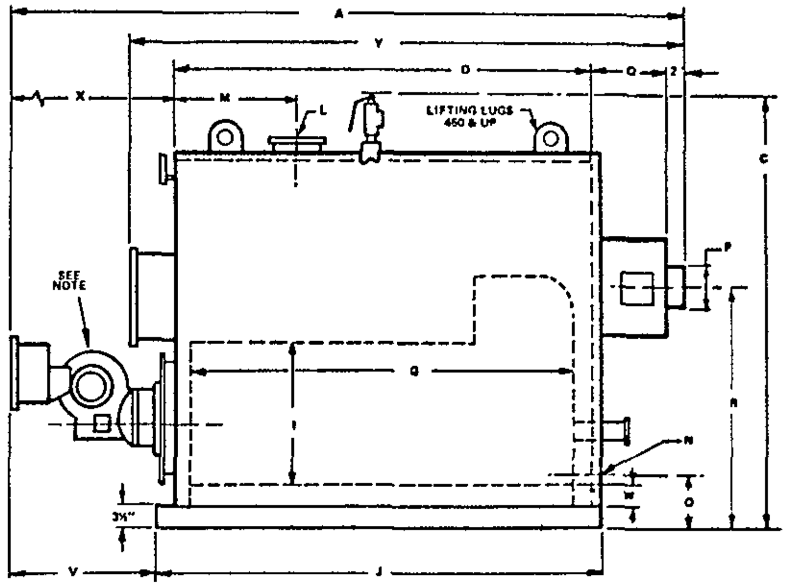

NOTE "Upright" type burner illustrated Sizes 275 thru 675 use an "SM" type burner

	BOILER NUMBER 36-45-	63	78	92	107	127	154	182	209	239	275	309	343	450	562	675
A	OVERALL LENGTH	68½	74½	80½	86½	80½	89½	97	106	96½	107½	115	122½	137	158	193
B	OVERALL WIDTH	39	39	39	39	43	43	43	43	49½	49½	49½	49½	64½	64½	64½
C	OVERALL HEIGHT	61	61	61½	61½	68½	68½	68½	70	78½	78½	78½	82	84½	85½	88
D	SHELL LENGTH	30½	36½	42½	48½	42½	50½	58	67	56½	64½	72	79½	91	112	132
E	SHELL WIDTH INSIDE	28½	28½	28½	28½	32½	32½	32½	32½	39½	39½	39½	39½	48	48	48
F	SHELL HEIGHT	55	55	55	55	62	62	62	62	71	71	71	71	78	78	78
G	FIREBOX LENGTH	25½	31½	37½	43½	37½	45	53	62	51	59	66½	74	84½	105½	125½
H	FIREBOX WIDTH	24	24	24	24	28	28	28	28	35	35	35	35	42	42	42
I	FIREBOX HEIGHT	15½	15½	15½	15½	20	20	20	20	25	25	25	25	33	33	33
J	BASE LENGTH	33½	39½	45½	51½	45½	53½	61½	70	59½	67½	75	82½	94	115	135
K	BASE WIDTH	32½	32½	32½	32½	37	37	37	37	44	44	44	44	54	54	54
L	SUPPLY SIZE	4"S	4"S	4"S	4"S	4"S	4"S	4"S	4"S	6"F	6"F	6"F	6"F	8"F	8"F	8"F
M	SUPPLY LOCATION	10	12	14	16	14	17	19	22	16	18	20	24	27	38	48
N	RETURN SIZE	3"	3"	3"	3"	4"	4"	4"	4"	4"	4"	4"	4"	4"	4"	4"
O	RETURN LOCATION	7	7	7	7	7½	7½	7½	7½	7½	7½	7½	7½	7½	7½	7½
P	SMOKE OUTLET DIA	6	6	6	6	8	8	8	8	10	10	10	10	16	16	18
Q	SMOKE BOX DEPTH	8	8	8	8	8	8	8	8	9	9	9	9	12	12	12
R	SMOKE OUTLET HEIGHT	36	36	36	36	41	41	41	41	47½	47½	47½	47½	52	52	52
S	SUPPLY HEIGHT	56	56	56	56	63	63	63	63	75½	75½	75½	75½	83	83	83
T	WASHOUT CONN HT.	47	47	47	47	54	54	54	54	62	62	62	62	67	67	67
U	NORMAL WATERLINE	46	46	46	46	53	53	53	53	61	61	61	61	65	65	65
V	BURNER TO BASE	26	26	26	26	26	27	27	27	27	30	30	30	30	30*	45
W	REFRACTORY THICKNS	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
X	TUBE PULL CLEARANCE	31½	37½	43½	49½	43½	51½	59	68	57½	65½	73	80½	92	113	133
Y	BOILER ONLY LENGTH	47¾	53¾	59¾	65¾	60¾	68¾	76¼	85¼	79¾	84¾	93¼	100¾	115¼	136¼	156¼

* - 45" FOR 6 3 BURNER

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7. DOMESTIC HOT WATER SUBSYSTEM

7.1 REQUIREMENTS

A review of the hot water supply requirements was conducted for the three types of system installations. In accordance with the latest methods developed by ASHRAE, the maximum hourly draw and required recovery rates were used to size the storage tanks and heaters shown in Table 7-1. It was assumed that 80 percent of the tank volume is usable. The storage tank sizes shown in the table are maximum and are paired with the minimum heater ratings. A larger heater rating results in a smaller storage tank; the maximum heater is an instantaneous type with no storage. Instantaneous heater sizes also are shown in Table 7-1. Heaters used in the actual system must include some additional capacity to make up storage tank and piping heat losses. The estimated tank losses shown in the table were based on the maximum tank sizes.

Figures 7-1 through 7-3 illustrate average daily hot water use profiles. Although system storage and heating capacity are sized for the maximum hourly draw and recovery (not included in the average use profile), these profiles are necessary in determining energy usage for system optimization. The profiles were developed from the 1976 edition of ASHRAE, and were used to arrive at the average daily consumptions of Table 7-1.

7.2 APPROACHES CONSIDERED

In optimizing the DHW subsystem prior to the proposal, a number of basic approaches were considered for the hot water supply. Of these, one subsystem features a dedicated solar collector, heat transport loops, and thermal energy

TABLE 7-1

HOT WATER SUPPLY SUBSYSTEM PARAMETERS

	Single Family	Multi-Family	Commercial*
Maximum draw, gph	72	144	24
Average daily consumption, gal	101	525	54
Maximum storage tank, gal	50	790	120
Minimum heater rate (no losses), Btu/hr	38,000	50,000	7100
Maximum heater rate (no storage), Btu/hr	85,400	171,000	28,500
Estimated hourly storage loss, Btu/hr	366	1600	577
Daily energy (including losses), Btu/day	92,622	474,150	58,674

*A factory or office building with 60 people using hot water for lavatory only.

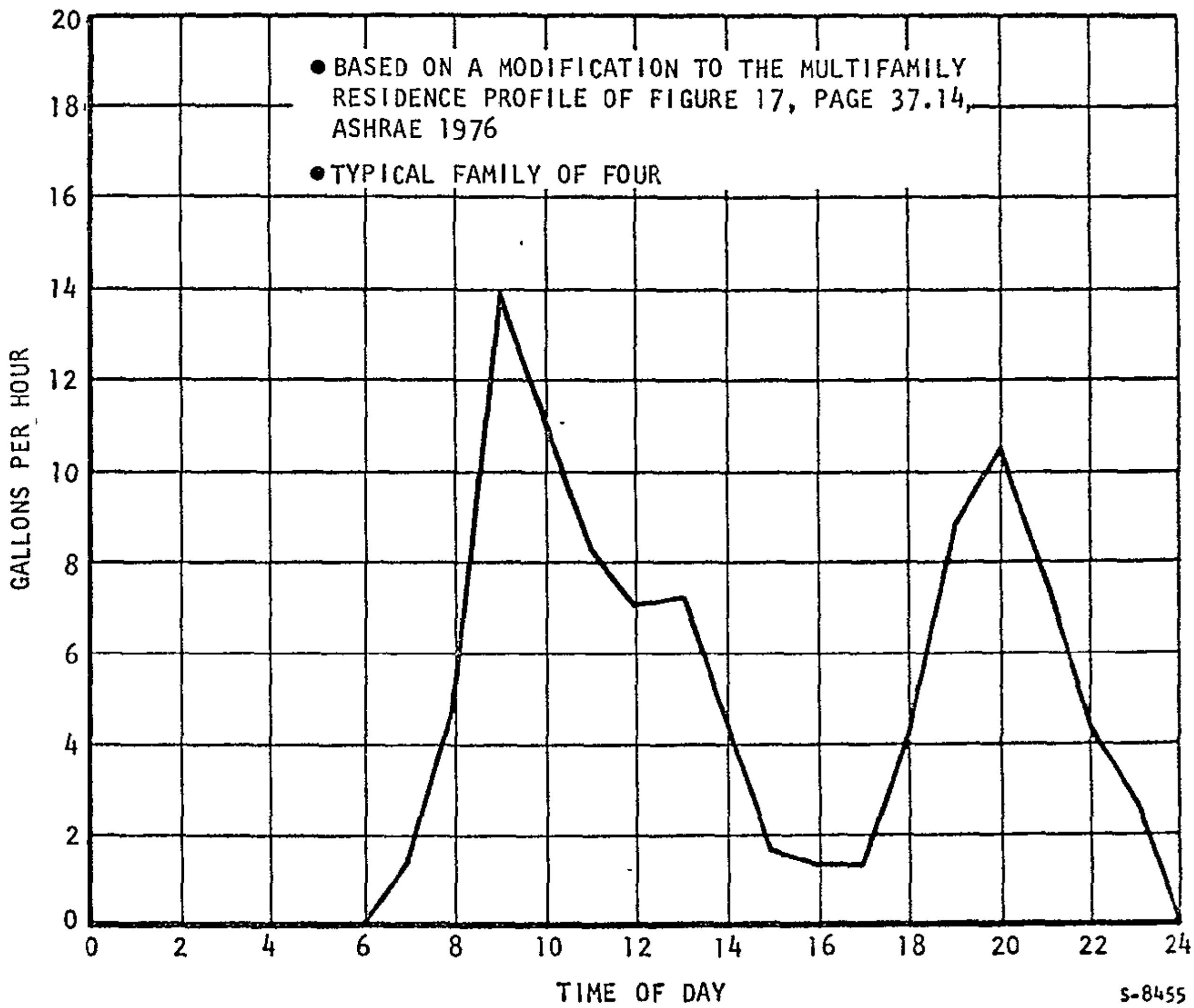

Figure 7-1. Single-Family Residence Hot Water Use

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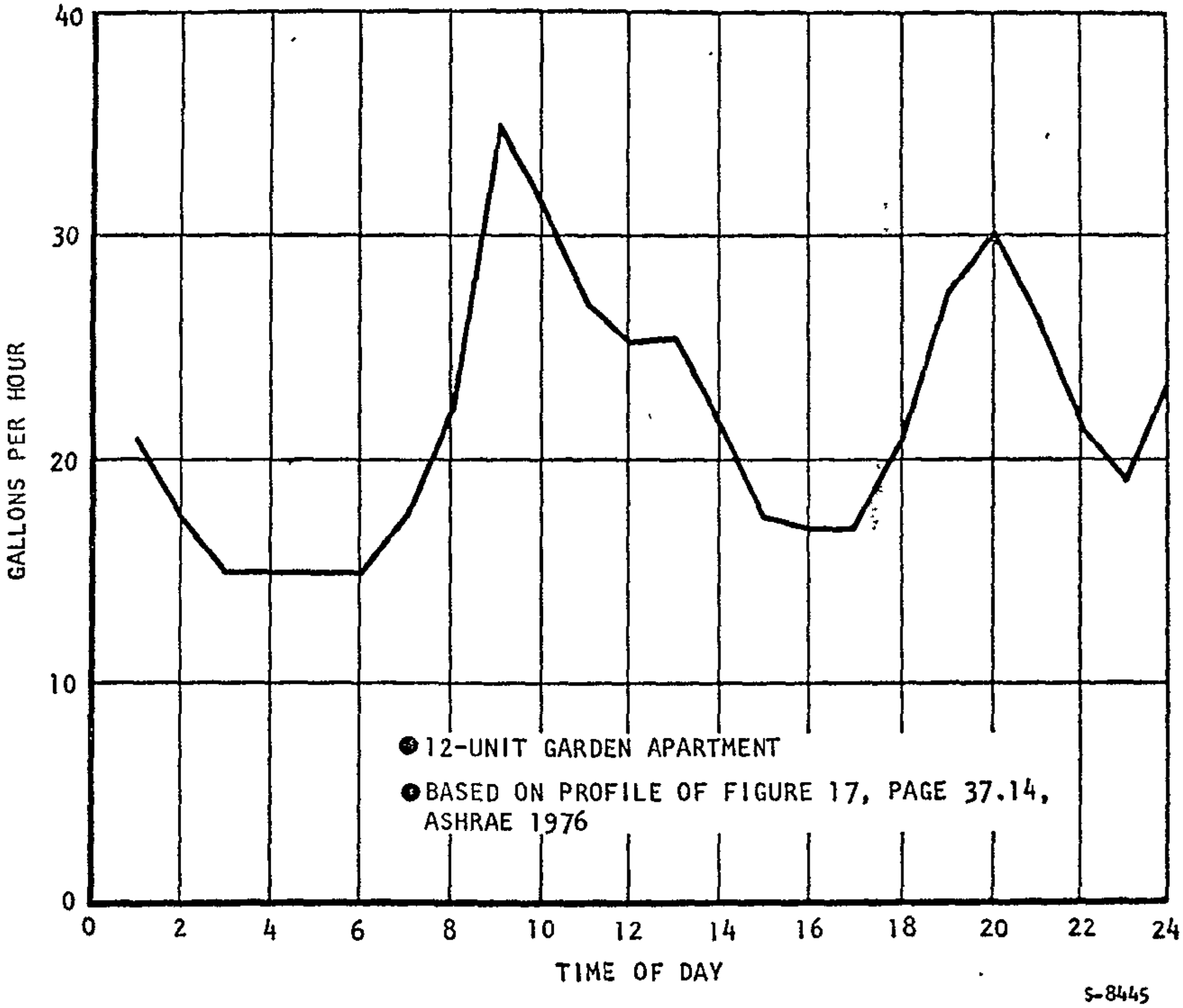


Figure 7-2. Multifamily Residence Hot Water Use

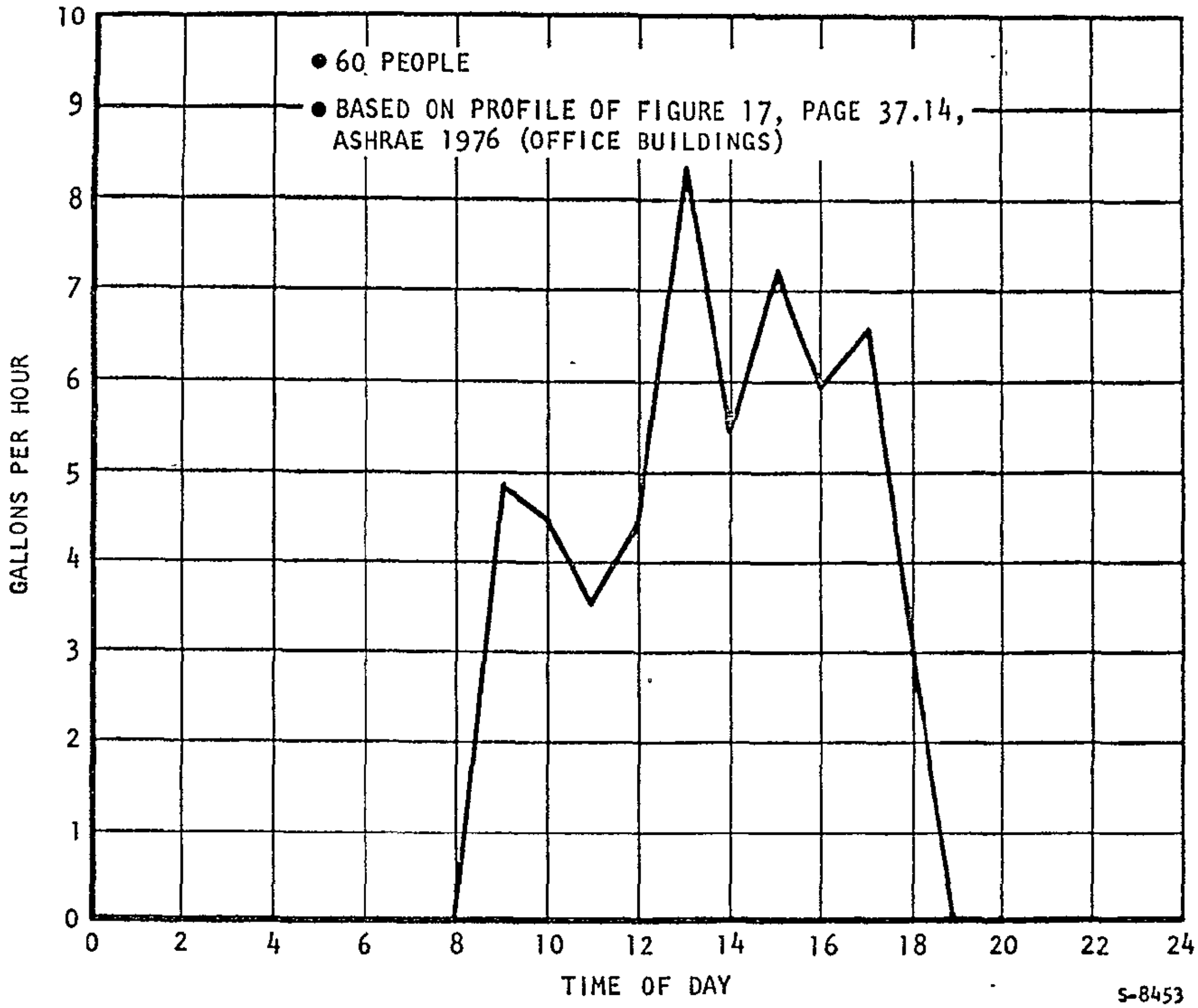


Figure 7-3. Commercial Application Hot Water Use

storage; another is a passive subsystem integrated with the main solar heating and cooling system for the building. On the basis of a life-cycle economic optimization, the integrated approach was selected. In view of the revised hot water demand model of paragraph 7.1, additional evaluation of these approaches is being conducted.

The passive integrated subsystem concept is considered the baseline. Schematics of this subsystem remain essentially the same (as described in para. 7.3). The optimization of this approach revolves primarily around the variation of DHW tanks and heater size. In the case of the commercial system, the hot water demand is relatively low and fairly constant throughout the day (Figure 7-3). For this subsystem, the use of an instantaneous makeup heater may be optimum.

For the approach using a dedicated solar collector loop, an additional thermal energy storage tank is employed to maximize use of solar energy during evening and early morning hours. The optimization of this approach centers on collector size and thermal energy storage requirements.

7.3 SUBSYSTEM DESCRIPTION

7.3.1 Baseline Subsystem

The schematics in Figure 7-4 illustrate the baseline hot water subsystem arrangements for the three applications. The single-family residence and commercial applications are schematically the same. A circulation pump is incorporated in the larger multifamily residence.

Water from the supply line is circulated through a coil around the main thermal energy storage tank and heated to a temperature approaching that of the tank. The coil is outside the tank for safety reasons. A mixing valve is provided to maintain the water temperature at the DHW tank inlet at a

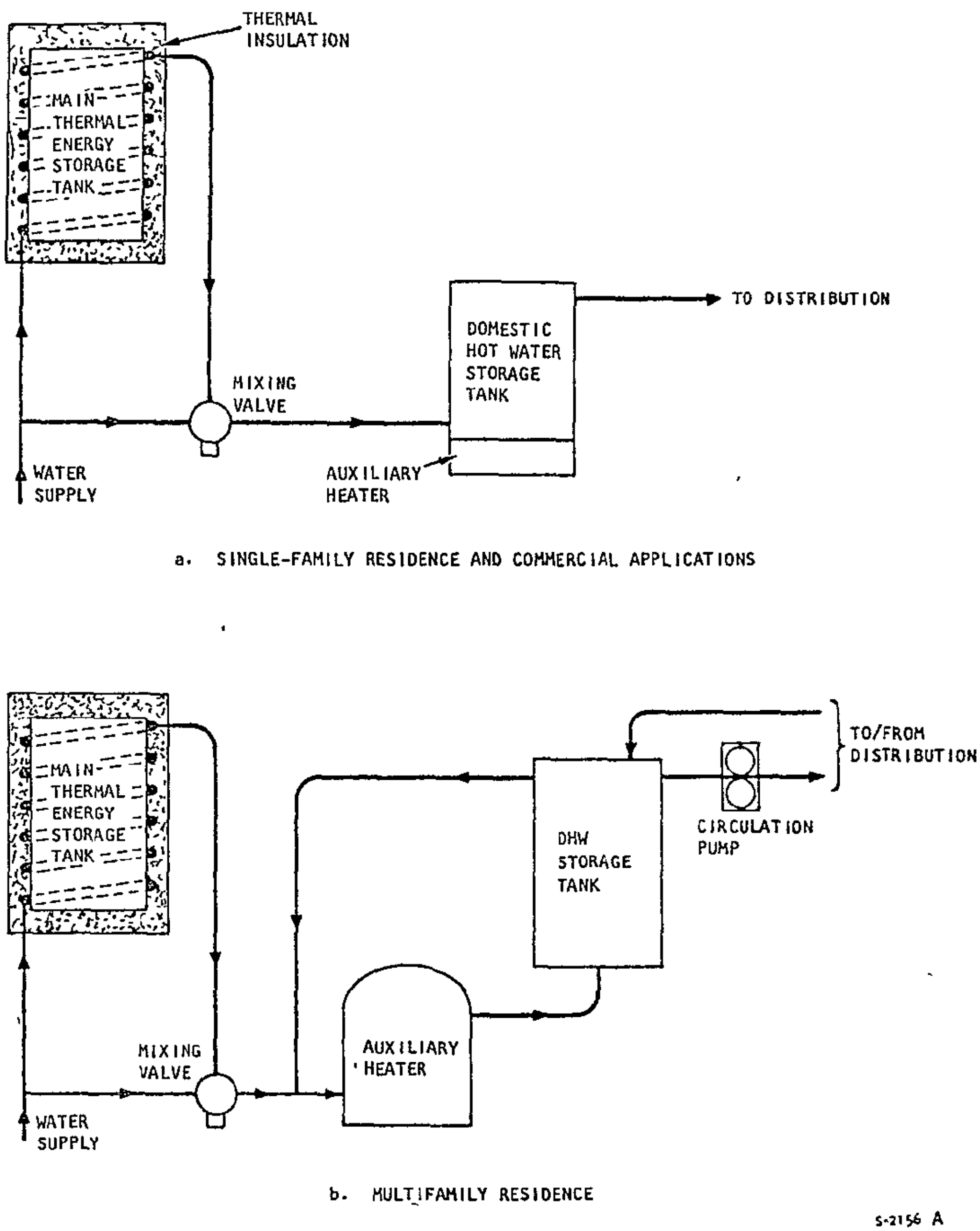


Figure 7-4. Baseline Hot Water Supply Subsystem Schematics

maximum temperature of 145°F (adjustable). When the main storage tank temperature is below 140°F, the auxiliary heater provided on the storage tank will automatically turn on and maintain the hot water supply at 140°F (adjustable).

For the multifamily residence, the higher withdrawal and recovery rates specified dictate the use of a larger DHW storage tank and auxiliary heater. A gravity recirculation loop is shown between these two components.

7.3.2 Alternate Subsystem

Solar heat energy is absorbed by a 50/50 water/glycol mixture which is circulated through the collector and then the interchanger (Figure 7-5). The interchanger provides efficient heat transfer to the water that is circulated through the energy storage tank. As hot water is drawn from the conventional DHW tank, the makeup water is added to the thermal energy storage tank in order to minimize DHW tank temperature changes.

7.4 SUBSYSTEM PERFORMANCE

Computerized subsystem optimization is currently being performed to update data generated previously. These data will be furnished to NASA when available.

7.5 EQUIPMENT SUMMARY

The equipment constituting the hot water supply subsystems is available as off-the-shelf commercial hardware. This equipment will meet or surpass the requirements of ASHRAE and the HUD Minimum Property Standards.

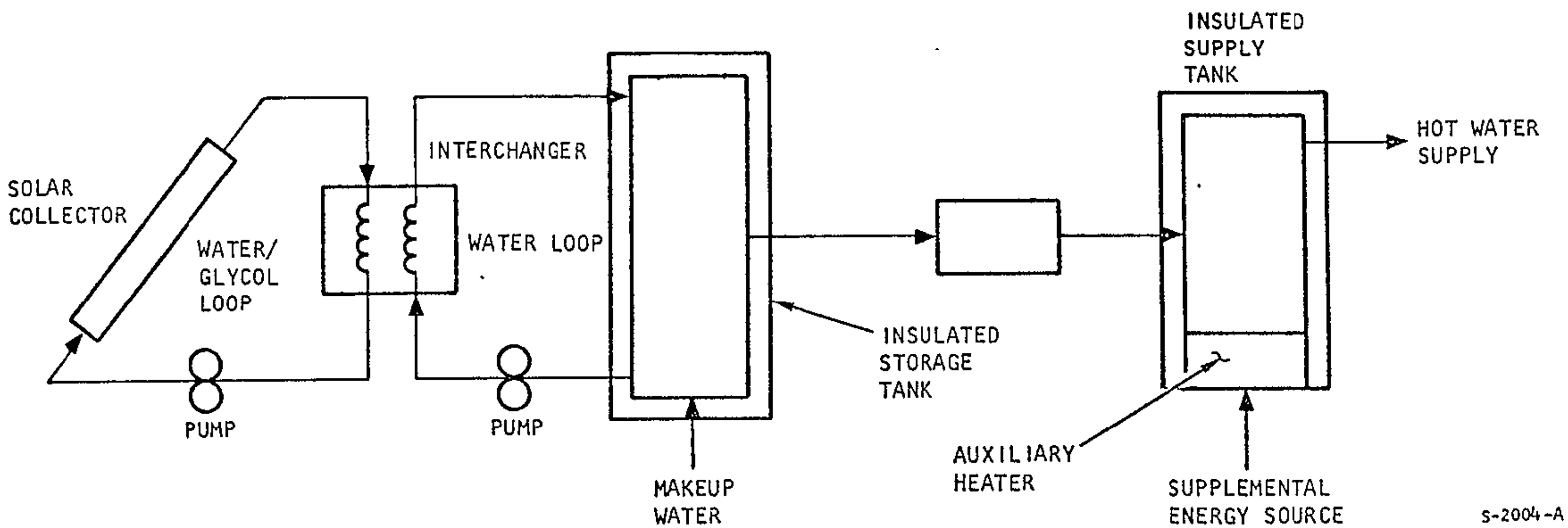


Figure 7-5. Alternate Hot Water Supply Subsystem Arrangement

8. SYSTEM CONTROL

8.1 GENERAL

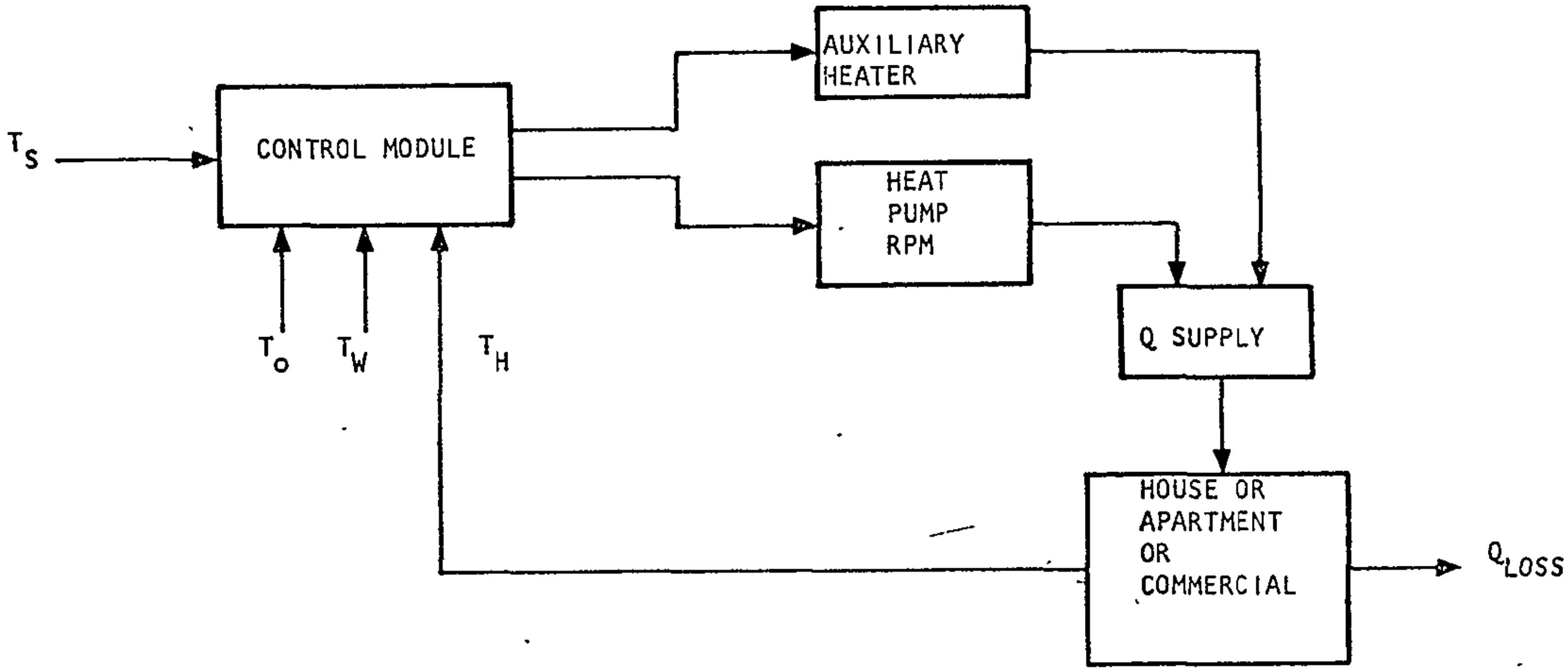
The control scheme for the heat pumps has been described in Section 5. The control module function is to gather information from the system temperature sensors, process this information, and issue appropriate commands to regulate the flow of heat into the conditioned space to achieve the desired heating effect with the minimum expense of utility power. Figure 8-1 shows the control module functional diagram.

Simulation of the dynamics of hardware performance as influenced by various sensing parameters, thermal capacity of the conditioned space, and heat loss from the structure, is necessary in order to ensure the stability of the control laws. The electronic control hardware then can be defined as a result of the simulation.

8.2 CONTROL MODULE MECHANIZATION

A control module block diagram is shown in Figure 8-2. T_H is the controlled variable that sums up the heating effect coming from the heat pump and/or the auxiliary heater. T_{ERROR} is the basic variable that computes the demand of heat to satisfy the desired temperature set point T_S. The temperature of water, T_W, in the storage tank acts as a safeguard in the heat pump operation to prevent compressor surge. T_W is also a measure of stored energy.

There are two basic outputs from the control module: the command to turn the auxiliary heater on or off, and the request of the heat pump to run



T_S = TEMPERATURE SETTING

T_O = OUTSIDE TEMPERATURE

T_W = WATER HEAT SOURCE TEMPERATURE

T_H = TEMPERATURE OF HEAT EFFECT (HOT WATER SUPPLY TO RESIDENCE)

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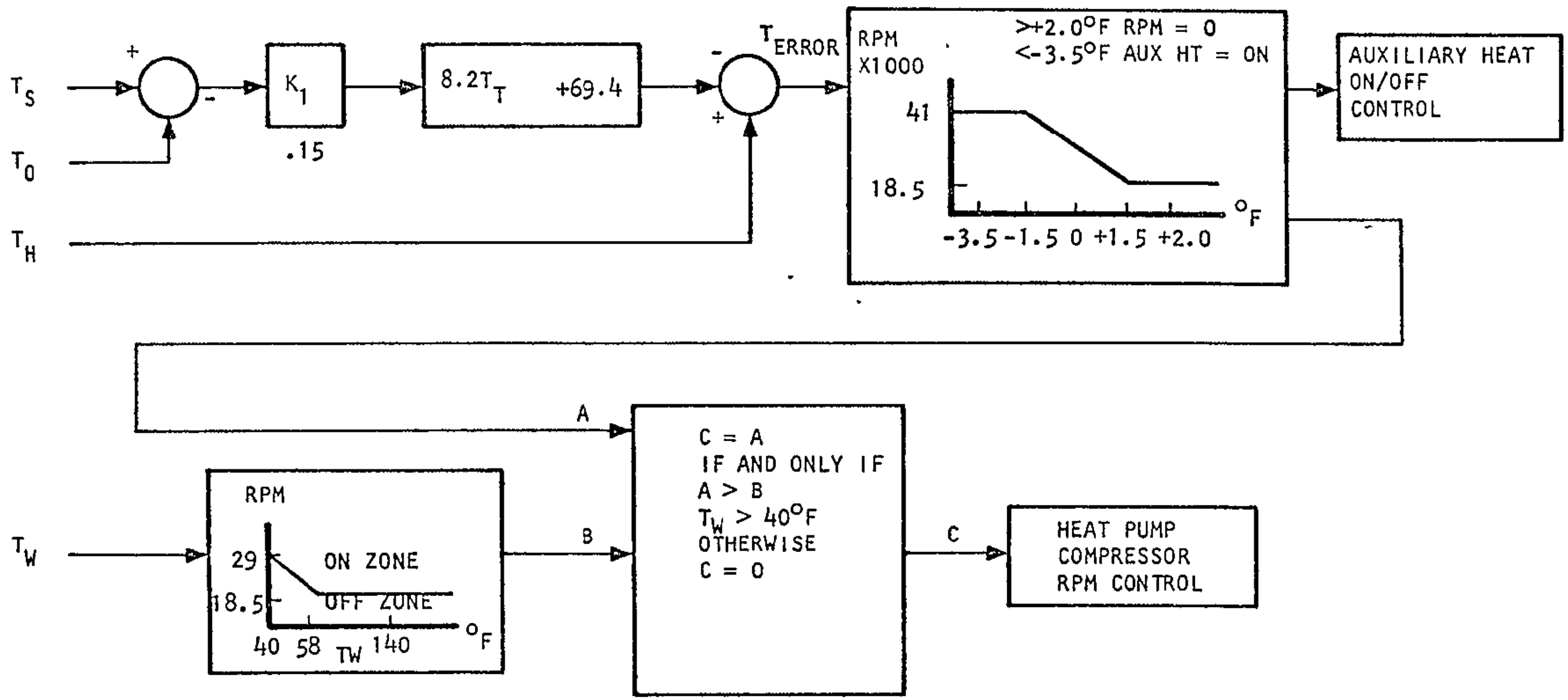
Figure 8-1. Control Module Functional Diagram



NOTE: 600 KBTUH HEAT PUMP SHOWN

S-8498

Figure 8-2. Control Module Block Diagram

at a certain rpm. Other inputs and outputs will be for the appropriate setup or shutdown of the heating systems.

Since the control laws are applicable in the three different capacity units, only one kind of control module is mechanized except for the outside temperature sensor. Primary design variations are the generation of heat pump rpm command vs T_{ERROR} input, and the function governing the relationship between rpm and T_W. This design approach will have favorable impact on control module cost and ease of maintenance.

8.2.1 Control Module Simulation

Simulation of the heating-only mode is currently being mechanized on a digital computer. The reaction of the heated space to heat input is modeled. The resulting temperature change is then fed back to the control module section to control the flow of heat into the heated space. Stability of control laws and first order effects can be studied effectively without the high cost of prototype testing. Direction of design improvements can be assessed by varying the appropriate parameters of the simulation model. Figure 8-3 shows the detail of the simulation setup for the 600 KBTUH unit. Data from this program will be available at the time of the preliminary design review.

8.2.2 Control Module Circuit Design

Hardware implementation of the control module is show in Figures 8-4, 8-5, and 8-6. Figure 8-4 shows circuits that develop rpm command for the heat pump operation and the on-off signal for the auxiliary heater as a function of input temperature variables. Temperature is measured by inexpensive thermistors housed in a suitable assembly commonly used in the industry.

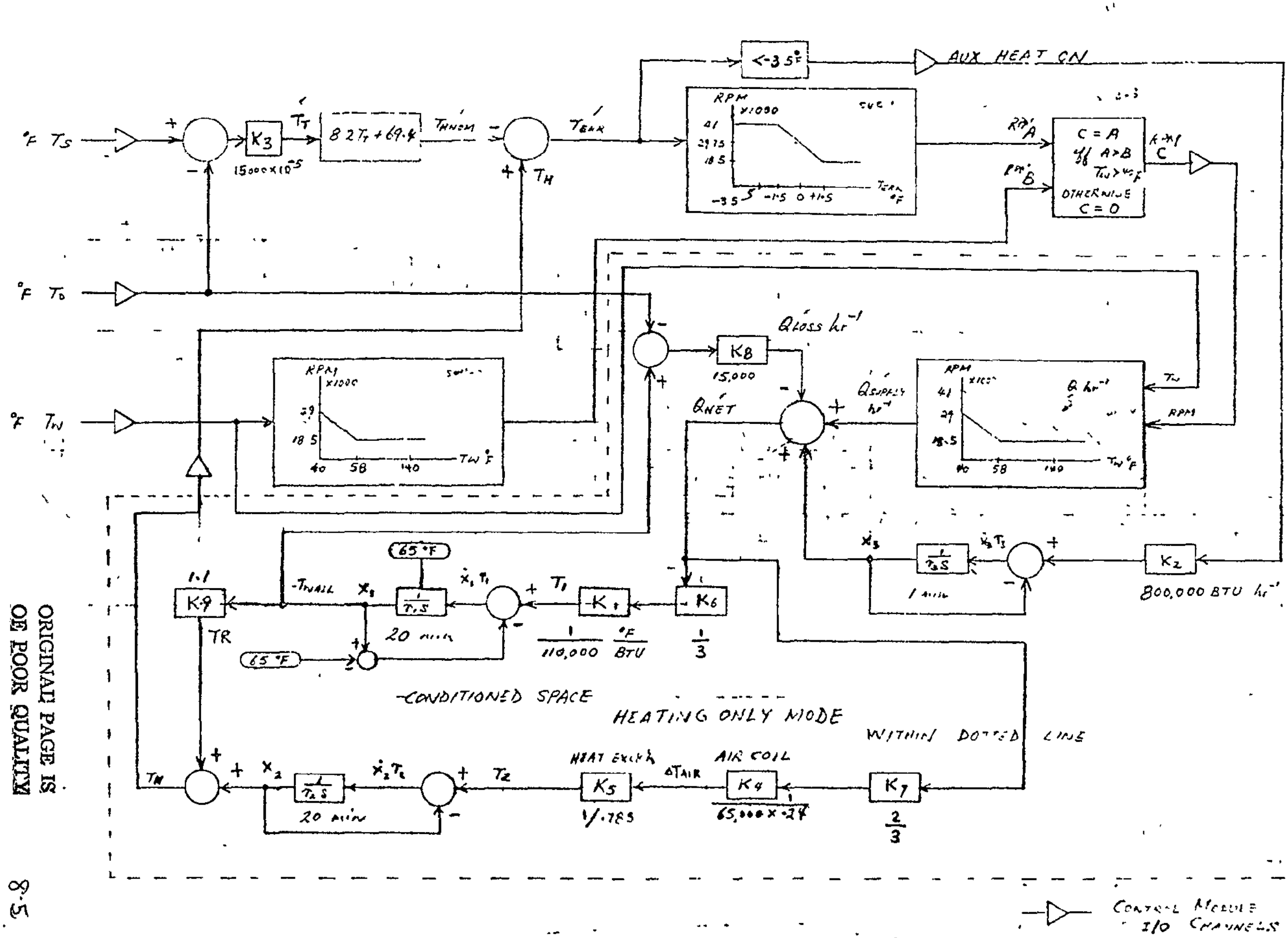


Figure 8-3. 600 KBTUH Heating Subsystem Simulation

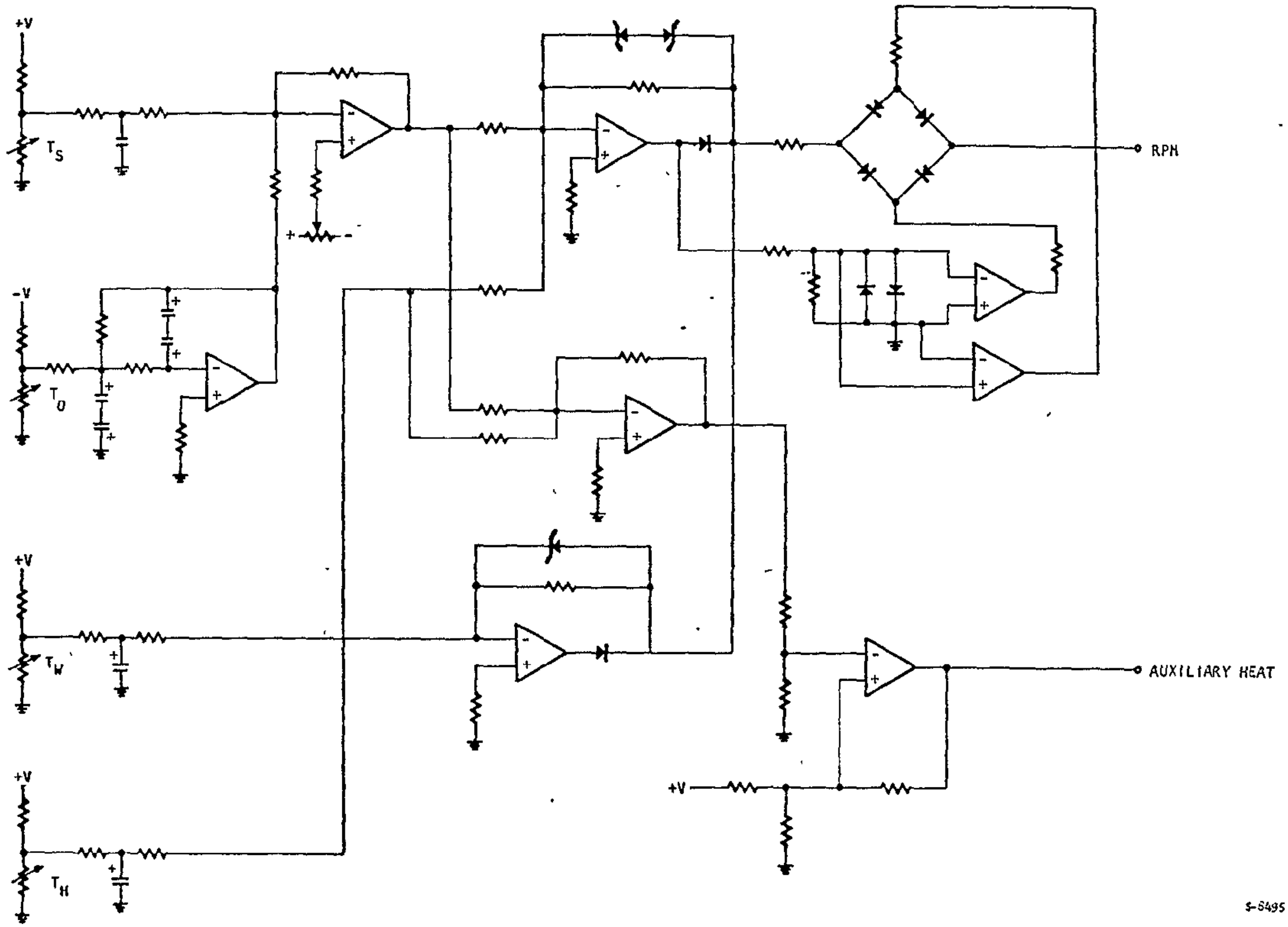


Figure 8-4. Compressor Rpm and Auxiliary Heater Command Circuitry

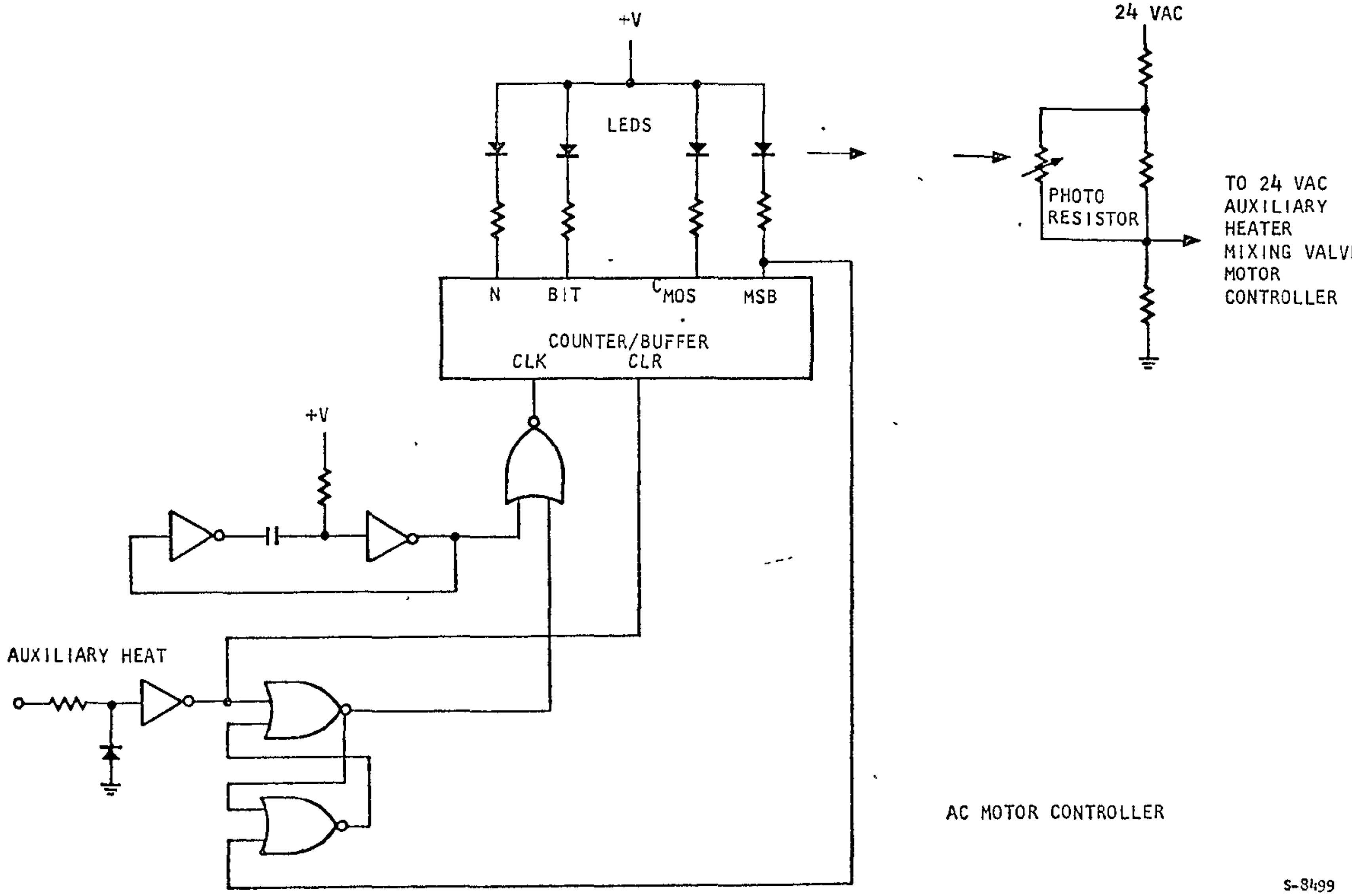


Figure 8-5. Auxiliary Boiler Mixing Valve Command

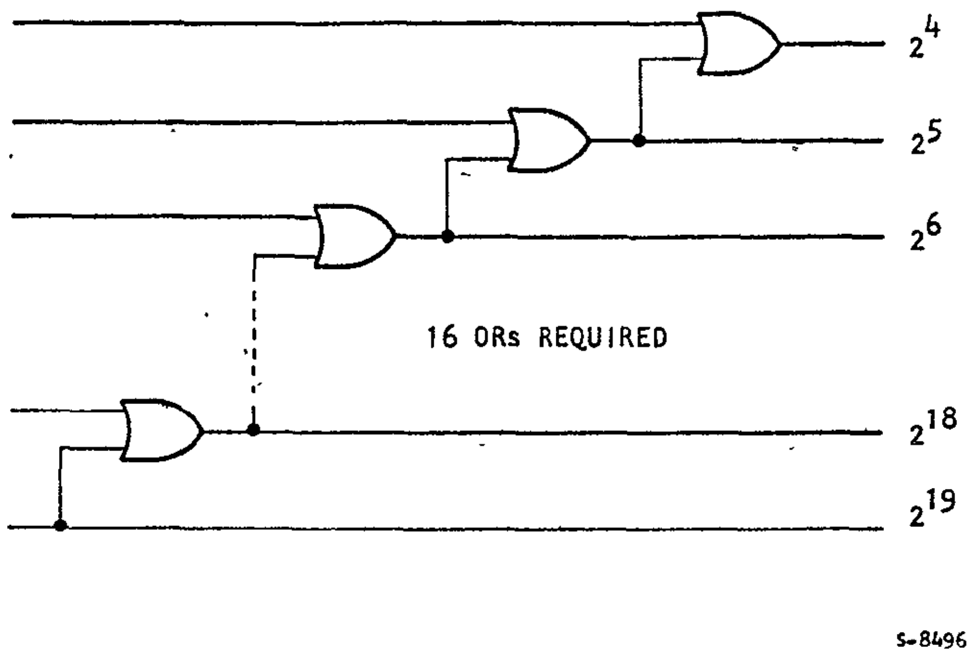


Figure 8-6. LED Driver-Buffer

Figure 8-5 shows the interface circuitry necessary to convert an auxiliary heater ON signal to a time rate of valve opening; the high temperature water available from the auxiliary boiler is thus mixed smoothly into the circulating water loop. High noise immunity logic elements such as CMOS will be used to generate a staircase signal, which in turn changes the resistance of the photo resistor by turning on successive LED's and thus varying the amount of light incident on the photo resistor. This approach also isolates the ac motor control circuit from the critical control module grounding system, thus allowing an electrically clean environment for the control circuitry.

Figure 8-6 shows the detail of the buffering between the electronic counter and the relatively higher power required by the LED's. An isolation interface may also be required between the rpm output and the high power PM motor control grounding because of high EMI generated in the motor control circuitry.

The circuit mechanization is heavily analog. Automotive type operational amplifiers will be used. CMOS logics are used to simplify power supply requirement. High noise immunity is an advantage in operating in the high power motor control environment.

9. TURBOMACHINE/MOTOR DETAIL DESIGN

9.1 INTRODUCTION

This section presents the results of the design effort expended to date on the heat pump turbomachines and motors. The basic design of all units is similar; size is the major difference between them. This design similarity will minimize assembly tooling and procedures.

It is important to note that the heating/cooling units represent a more difficult design problem than the heating-only units. Power balance, parasitic losses, critical speeds, and aerodynamic balance are more critical for the heating/cooling machines. Therefore, commonality of hardware dictates that these be designed first. Conversion to the heating-only configuration is a relatively simple task.

Detail design efforts are being expended in the following major areas for all sizes of machines.

- Motor

  Magnet cylinder
  Position/speed sensor
  Stator-connector

- Compressor and diffuser
- Turbine and nozzle
- Rotor assembly

  Bearings
  Critical speeds
  Seals

- Cooling and lubricant flow

The layout drawings presented reflect current design status. Details of the layouts could change as detailed analyses are completed.

Detail design efforts were concentrated on the small residential application machine. This unit was selected for initial work because it is the first one scheduled for delivery.

9.2 SINGLE-FAMILY RESIDENCE UNIT (3 TON/60 KBTUH)

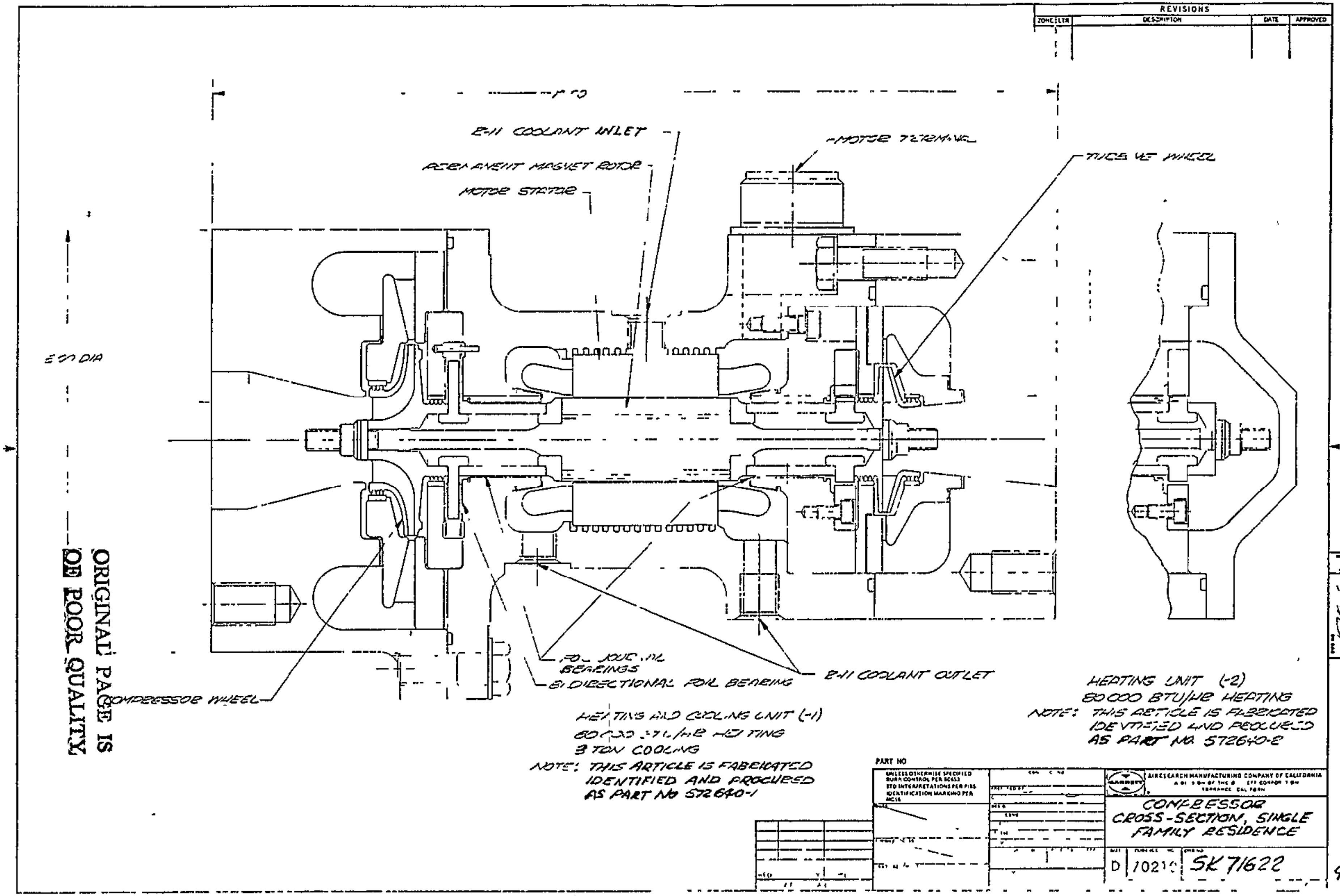
A cross-section of this machine is shown in SK71622; for reference, SK71125, from the AiResearch proposal, is also included. The following changes have been made in the design since the proposal:

- Incorporation of separate thrust and journal bearings as opposed to two conical bearings
- Incorporation of a Hall effect speed and position sensor instead of the slotted rotating shutter sensor
- A slightly longer motor resulting from detailed analysis

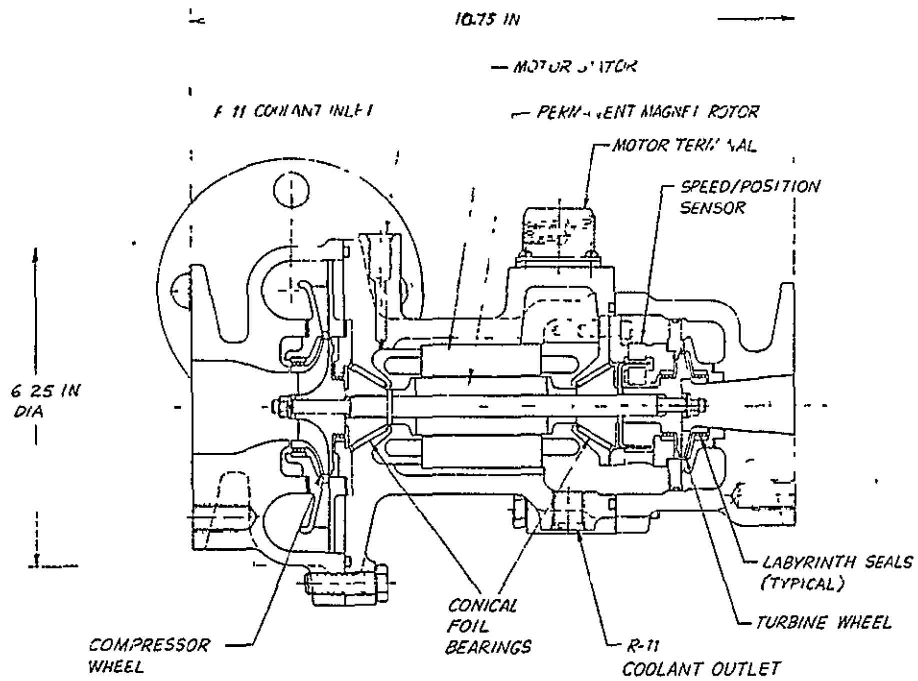
The new bearing configuration offers better load carrying capacity; also it provides design similiarily to the other sizes. The overall parasitic loss remains the same with the thrust-and-journal bearing arrangement because of (1) recent improvements in the bearing performance, and (2) use of a smaller position sensor, with lower parasitic drag.

A Hall effect-type sensor was selected because of simplicity and lower cost. Both sensor types will be tested in a bearing/motor test unit currently under development to assess the effectiveness of each type. It is felt that the Hall effect sensor will be superior in performance; it also is a simpler design, has less friction, and eliminates a difficult fabrication problem.

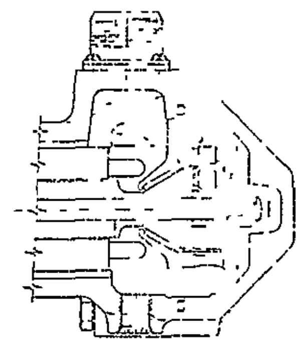
The change in motor size is the result of refinement of the problem statements and detailed design analysis.


REVISIONS
ZONE LTR DESCRIPTION DATE APPROVED
R-11 COOLANT INLET
PERMANENT MAGNET ROTOR
MOTOR STATOR
MOTOR TERMINAL
TURBINE WHEEL
DIA
FOIL JOURNAL BEARINGS
BI-DIRECTIONAL FOIL BEARING
R-11 COOLANT OUTLET
COMPRESSOR WHEEL
HEATING AND COOLING UNIT (-1)
80 000 BTU/HR HEATING
3 TON COOLING
NOTE: THIS ARTICLE IS FABRICATED
IDENTIFIED AND PROCURED
AS PART No 572640-1
HEATING UNIT (-2)
80 000 BTU/HR HEATING
NOTE: THIS ARTICLE IS FABRICATED
IDENTIFIED AND PROCURED
AS PART NO. 572640-2
PART NO
AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA
COMPRESSOR
CROSS-SECTION, SINGLE
FAMILY RESIDENCE
D 70210 SK 71622

LTR	REVISIONS DESCRIPTION	DATE	APPROVED



HEATING AND COOLING UNIT (-1)
80,000 BTU/HR-HEATING
3 TON - COOLING
NOTE THIS ARTICLE IS FABRICATED, IDENTIFIED AND PROCURED AS PART NO 572105-1



HEATING UNIT (-2)
80,000 BTU/HR-HEATING
NOTE THIS ARTICLE IS FABRICATED, IDENTIFIED AND PROCURED AS PART NO 572105-2

SK 71125

QTY REQD	ITEM NO	CODE IDENT	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION	SYM
			PARTS LIST		

UNLESS OTHERWISE SPECIFIED, BURR CONTROL PER SC663
IDENTIFICATION MARKING PER MC16
STD INTERPRETATIONS PER PI65
HEAT TREATMENT
HARDNESS AND SPEC

REQD | NEXT ASSY | USED ON
APPLICATION

A RESEARCH MANUFACTURING COMPANY OF CALIFORNIA

COMPRESSOR CROSS SECTION,
SINGLE FAMILY RESIDENCE

SIZE	CODE IDENT NO	DWG NO
D	70210	SK 71125

SCALE | SHEET OF

9-4

9.2.1 Layout Description

The major element of the machine is the rotating assembly, which in this case consists of the permanent magnet (PM) motor rotor, the compressor wheel, the thrust bearing disk, two journal bearings, two retaining rings for balancing, the speed and position sensing disk, and the turbine wheel. A solid shaft made from Inconel 713 extends the full length of the rotating assembly. Brazed to the shaft is a slotted cylinder of permeable steel, which retains the magnets for the motor rotor. The complete assembly is clamped together with some individual components brazed or shrunk onto the shaft. Nuts on either end of the shaft lock the total assembly together.

The center housing is made of aluminum and contains the motor stator. The compressor scroll and turbine torus are aluminum; they bolt to the center housing. Sandwiched between torus, scroll, and center housings are the turbine nozzle assembly, the compressor diffuser assembly, and the labyrinth seals to prevent leakage around the wheels. Shims are provided at both ends of the machine for proper positioning of the nozzle and diffuser assemblies in relation to the rotating assembly. O-ring seals are provided to prevent any external leakage. Because of the small mass of the rotating assembly, no special provision is included in the design for containment.

The speed/position sensing ring is located near the turbine end of the unit and surrounds a slotted ferrite rotating disk. The sensing ring consists of three Hall effect sensors, which are located 120 deg apart around the ring. As the disk rotates, signals are generated in the Hall effect sensors and fed to the control system for proper motor control. Self-acting foil segment journal and thrust bearings are contained in the housing and are held in position by rings and pins respectively.

Orifices are located in the center housing for metering a small flow of refrigerant for motor and bearing cooling. The refrigerant is bled into the motor cavity from the condenser outlet and is returned to the evaporator inlet. The pressure in the motor cavity is maintained slightly above compressor inlet pressure.

The construction and assembly of the heating-only machine are identical to those of the heating/cooling unit, except that the turbine wheel is replaced with an equivalent weight disk and the torus and nozzle assembly are replaced with a close-off plate.

9.2.2 Performance Requirements

The performance requirements for the three elements that constitute the heating/cooling and heating-only machines are shown in Table 9-1. The values shown are the result of system analysis and represent design point conditions for the machine rather than for the complete heat pump.

Preliminary design of the compressor and turbine were based on these data. Performance maps were developed subsequently and used in heat pump subsystem analyses to assure performance over a wide range of conditions. These maps are shown in Figures 9-1 through 9-3.

9.2.3 Detailed Aerodynamic Design

The results of the detailed aerodynamic design follow. The principal dimensions and blade shape of the compressor impeller wheel are shown in Figure 9-4.

The turbine blade principal dimensions and coordinates are listed in Table 9-2. The blade shape is shown in Figure 9-5.

A cross-sectional view of the compressor, impeller, and turbine wheels is shown in Figure 9-6.

TABLE 9-1

3-TON TURBOCOMPRESSOR

	Cooling Mode	Heating Mode
Compressor		
Fluid type	R-11	R-11
Inlet pressure, psia	8.6	9.04
Inlet temperature, °F	55	51.2
Inlet H, Btu/lb	99.31	98.79
Outlet pressure, psia	21.78	35.59
Outlet H, Actual, Btu/lb	109.05	115.10
ΔH adiabatic, Btu/lb	7.01	10.93
ΔH actual, Btu/lb	9.74	16.31
Volume flow, inlet, cfs	0.55	0.86
Weight flow, lb/sec	0.122	0.20
Speed, rpm	60,000(constant)	82,000
Theoretical efficiency, percent	72	67
Power at wheel, kw	1.25	3.42
Turbine		
Inlet pressure, psia	86	NA
Inlet temperature, °F	190	
Inlet H, Btu/lb	115.38	
Outlet pressure, psia	21.78	
Outlet H, Btu/lb	106.64	
ΔH adiabatic, Btu/lb	10.93	
ΔH actual, Btu/lb	8.74	
Volume flow, inlet, cfs	0.3833	
Weight flow, inlet, lb/sec	0.150	
Speed, rpm	60,000(constant)	
Theoretical efficiency, percent	80	
Power at wheel, kw	1.316	
Motor		
Maximum speed, rpm	60,000(constant)	82,600
Minimum speed, rpm	NA	36,500
Power at maximum speed, kw	1.4*	3.42**
Power at minimum speed, kw	1.4*	NA
Growth potential, power, percent	NA	20
Growth potential, speed, percent	10	10
Maximum overspeed, rpm	NA	90,860
Pressure, cavity, psia	9.6	9.6
Temperature, cavity, °F	100	150
Viscosity, μ, Reynolds	1.64×10^{-9}	1.75×10^{-9}
Motor efficiency, percent	NA	91.18
Bearings/lubricant, types	Foil/R-11/vapor	Foil/R-11/vapor
Cooling flow, lb/sec (approx)	0.0024	0.0063
Type motor	PM Samco	PM Samco
No-load power	TBD	NA
Maximum design speed, rpm	NA	100,000
General		
Design life, yr	20	20
No. of starts, lifetime	60,000	60,000
Service hours, lifetime	75,000	50,000
Type control	on/off	Modulating

* Power delivered to the compressor assuming 5 percent mechanical losses.

** Power delivered to the compressor assuming no mechanical losses.

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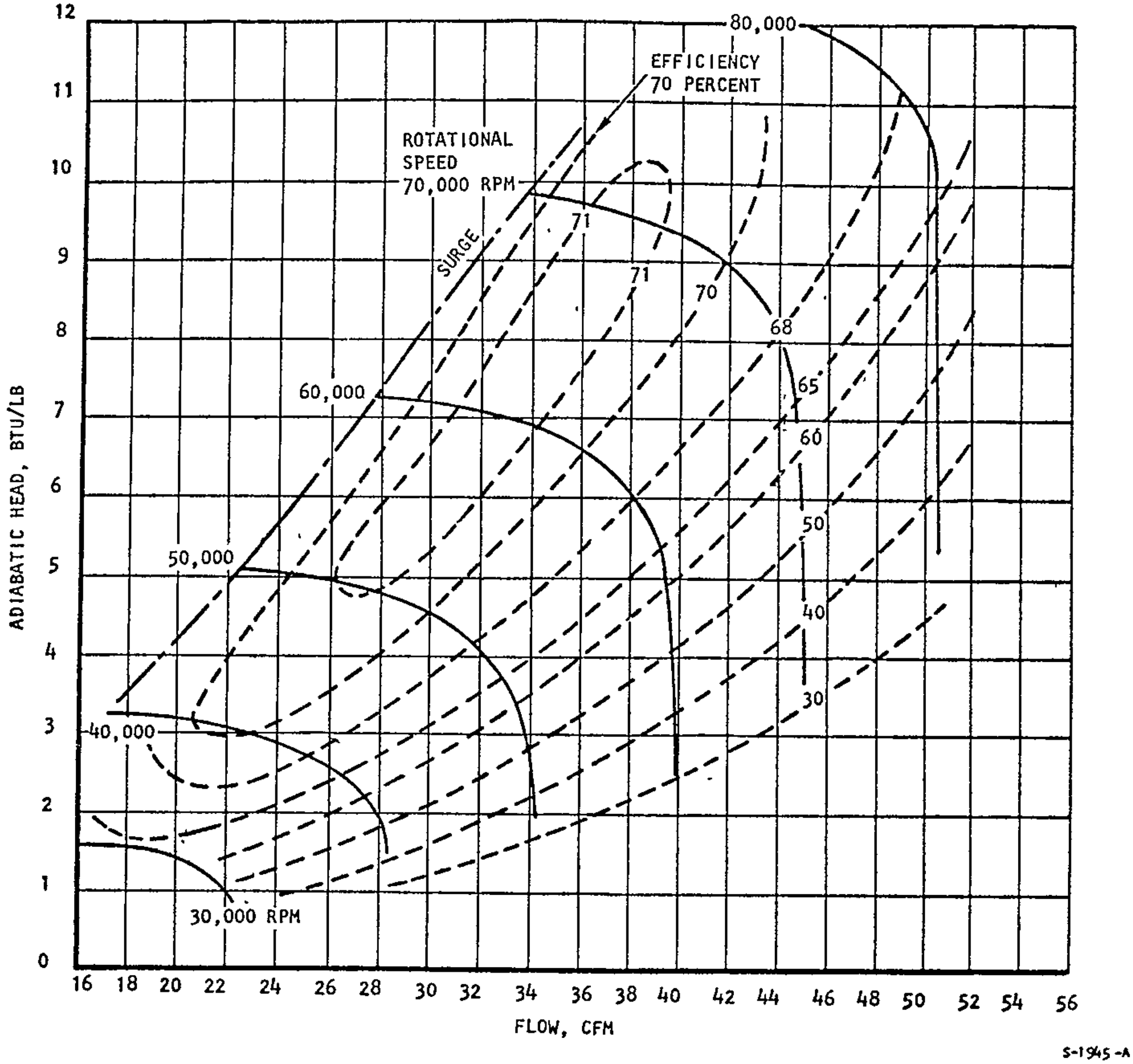


Figure 9-1. Compressor Map (3-Ton/80,000-Btu/hr Unit)

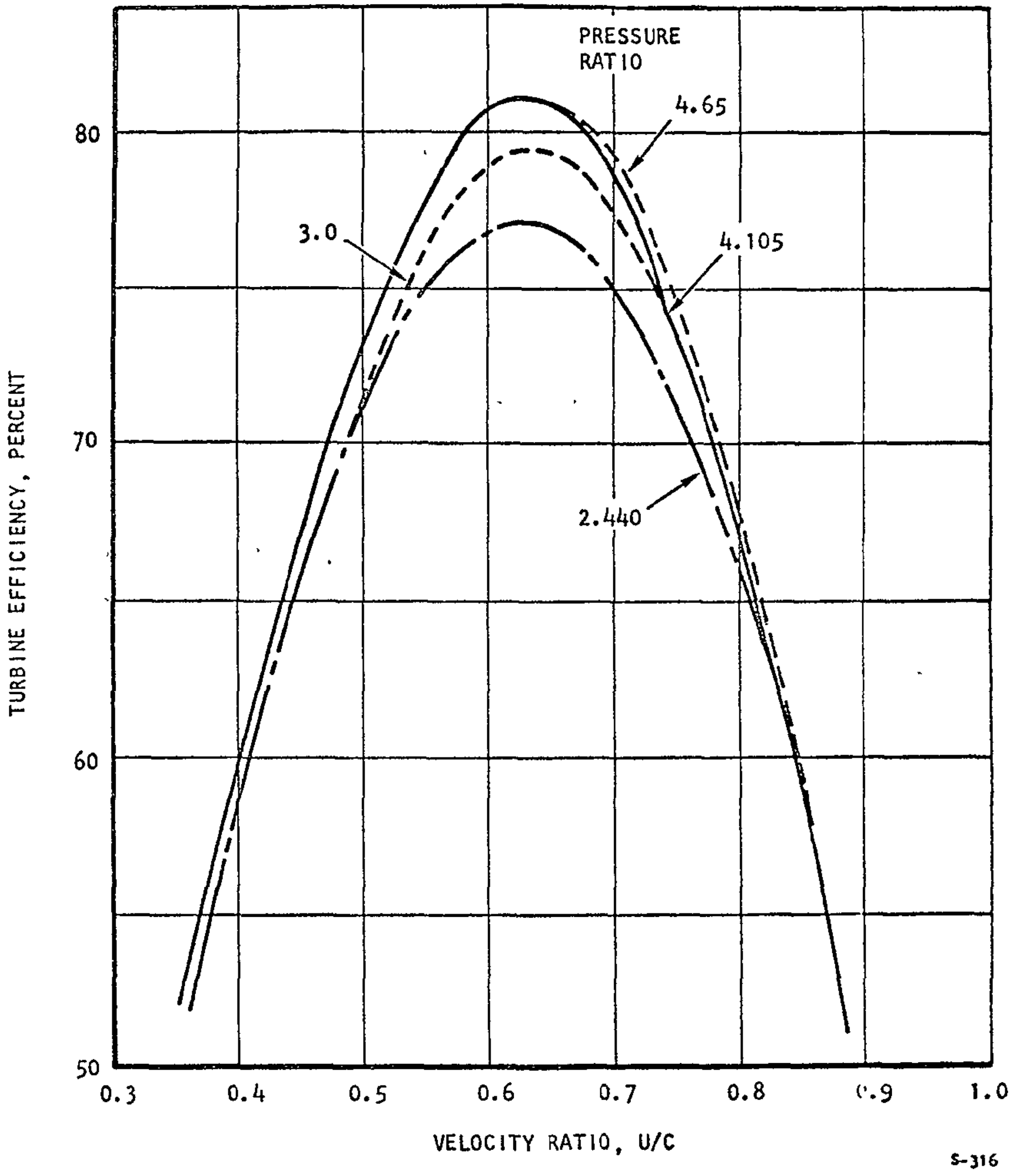


Figure 9-2. Turbine Efficiency of 3-Ton Unit

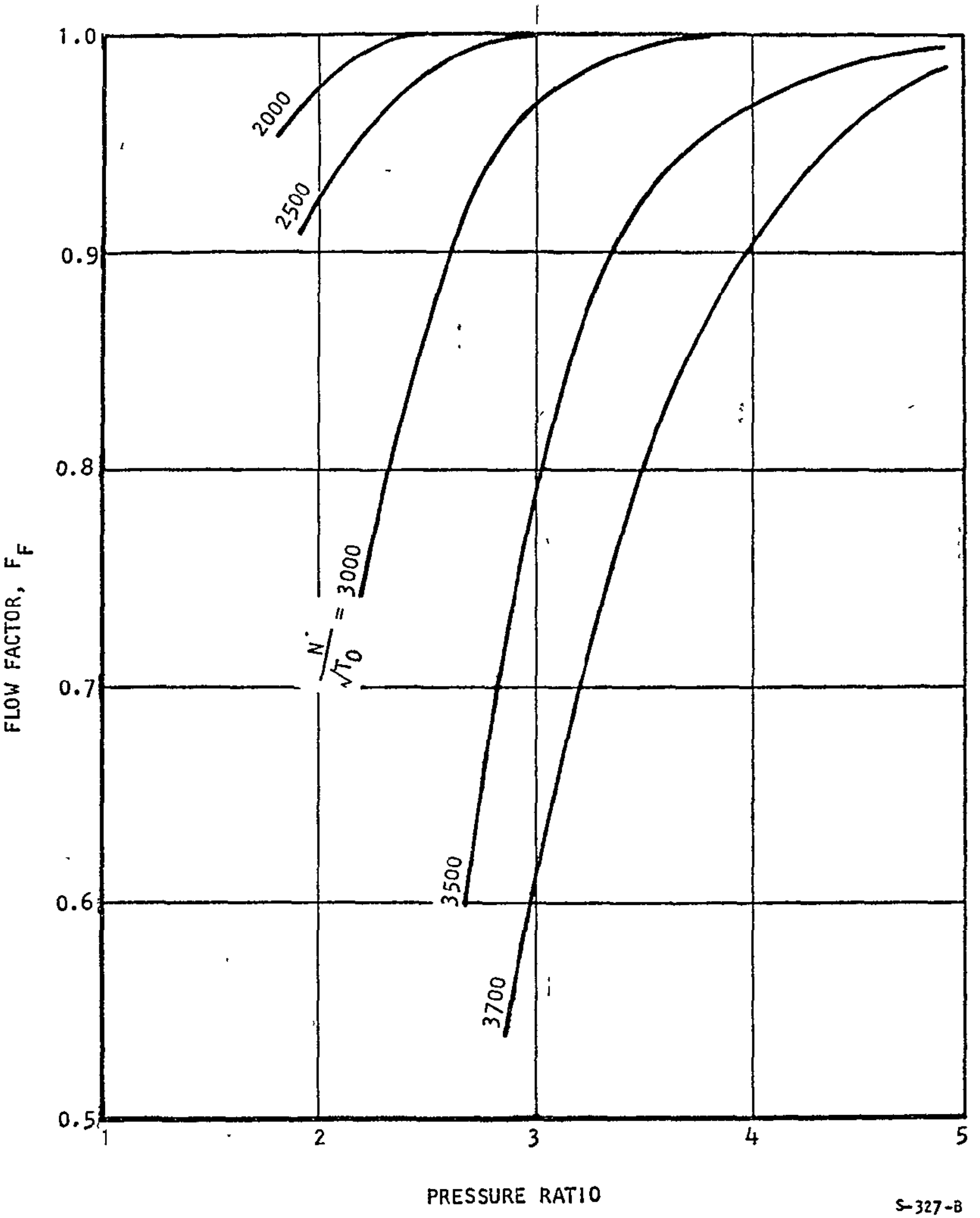
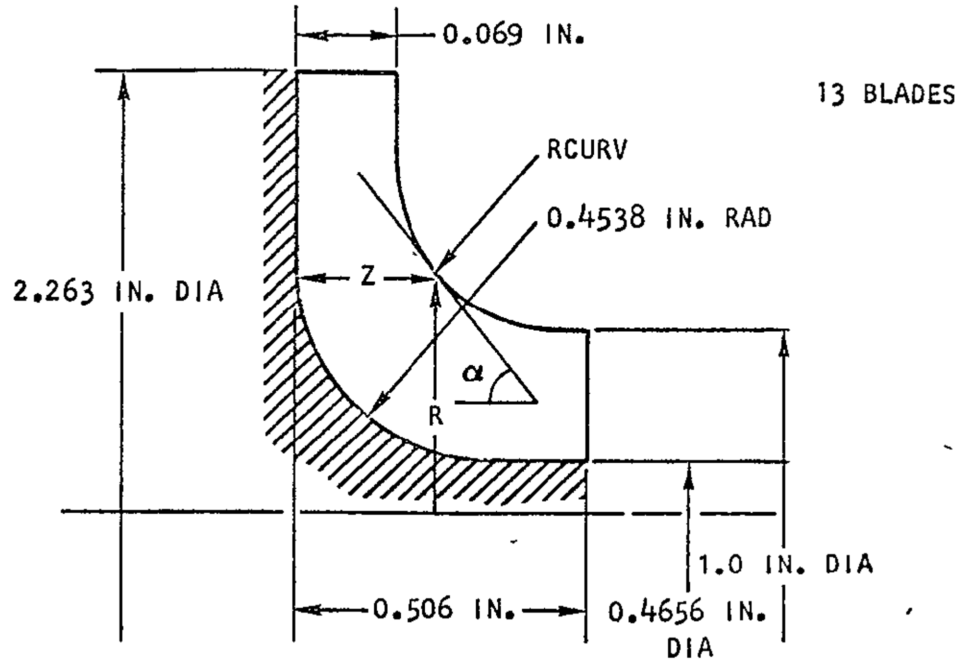


Figure 9-3. Turbine Flow Coefficient (3-Ton Unit)



I	ZETA	Z	R	SLOPE	RCURV
1	1.173440	.718604	.506315	2.629828	13.900325
2	1.284516	.598635	.501579	1.725471	4.704728
3	1.395591	.506000	.500000	.000014	2.115455
4	1.506667	.433087	.501579	-2.783077	1.098025
5	1.617742	.374671	.506315	-6.862634	.631018
6	1.728818	.327149	.514209	-12.425165	.398257
7	1.839893	.287970	.525260	-19.499328	.278935
8	1.950969	.255291	.539469	-27.822167	.221001
9	2.062044	.227749	.556835	-36.786262	.200952
10	2.173120	.204322	.577359	-45.596526	.208941
11	2.284195	.184228	.601040	-53.575252	.242568
12	2.395270	.166865	.627879	-60.366559	.304004
13	2.506346	.151758	.657875	-65.921532	.398564
14	2.617421	.138534	.691029	-70.369177	.534032
15	2.728497	.126691	.727340	-73.899858	.720395
16	2.839572	.116588	.766809	-76.700778	.969611
17	2.950648	.107426	.809435	-78.931288	1.296703
18	3.061723	.099244	.855219	-80.718542	1.717926
19	3.172799	.091905	.904160	-82.161043	2.252973
20	3.283874	.085299	.956259	-83.334182	2.924203
21	3.394950	.079330	1.011515	-84.295512	3.757094
22	3.506025	.073920	1.069929	-85.089072	4.780508
23	3.617100	.069000	1.131500	-85.748739	6.026966
24	3.728176	.064513	1.196229	-86.300739	7.532941
25	3.839251	.060411	1.264115	-86.765526	9.339160

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Figure 9-4. Compressor Impeller Blade Coordinates

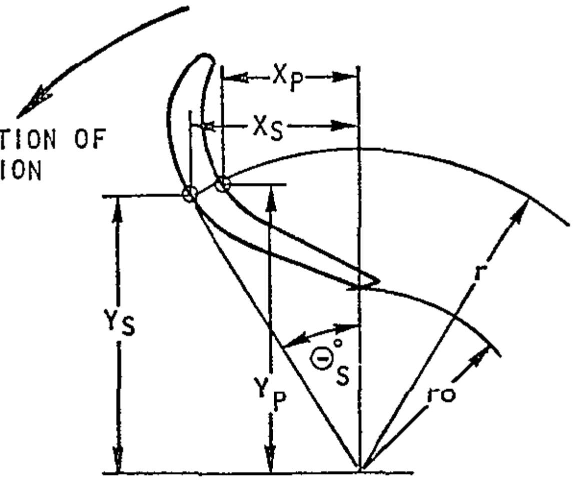


TABLE 9-2

TURBINE WHEEL BUCKET COORDINATES

r/r_o	r	θ_s^o	X_s	Y_s	θ_p^o	X_p	Y_p
1.0	0.43075	0	0	0.43075	-3.194	-0.02400	0.430080875
1.1	0.473825	10.622815	0.0873462	0.4657046	7.428815	0.061263	0.46984784
1.2	0.5169	18.42053	0.1633347	0.4904155	15.22653	0.1357565	0.49875422
1.3	0.559975	23.99137	0.2276853	0.5115969	20.79737	0.1988270	0.523488136
1.4	0.60305	27.78233	0.2810900	0.5335333	24.58833	0.25092645	0.5483660
1.5	0.646125	30.128354	0.3243152	0.5588355	26.934354	0.2926748	0.57603730
1.6	0.6892	31.2907	0.357957	0.58895113	28.0967	0.3245864	0.607980532
1.65	0.7107375	31.49359	0.3712915	0.60604488	28.29959	0.33694779	0.62579068
1.7	0.732275	31.47366	0.3823256	0.6245430	28.27966	0.346934034	0.64487476
1.8	0.77535	30.840	0.3974773	0.6657172	27.646	0.35976812	0.68682933
1.9	0.818425	29.520	0.4032604	0.71218014	26.326	0.3629534	0.73354228
2.0	0.8615	27.620	0.399396	0.7633250	24.426	0.35624545	0.7843924

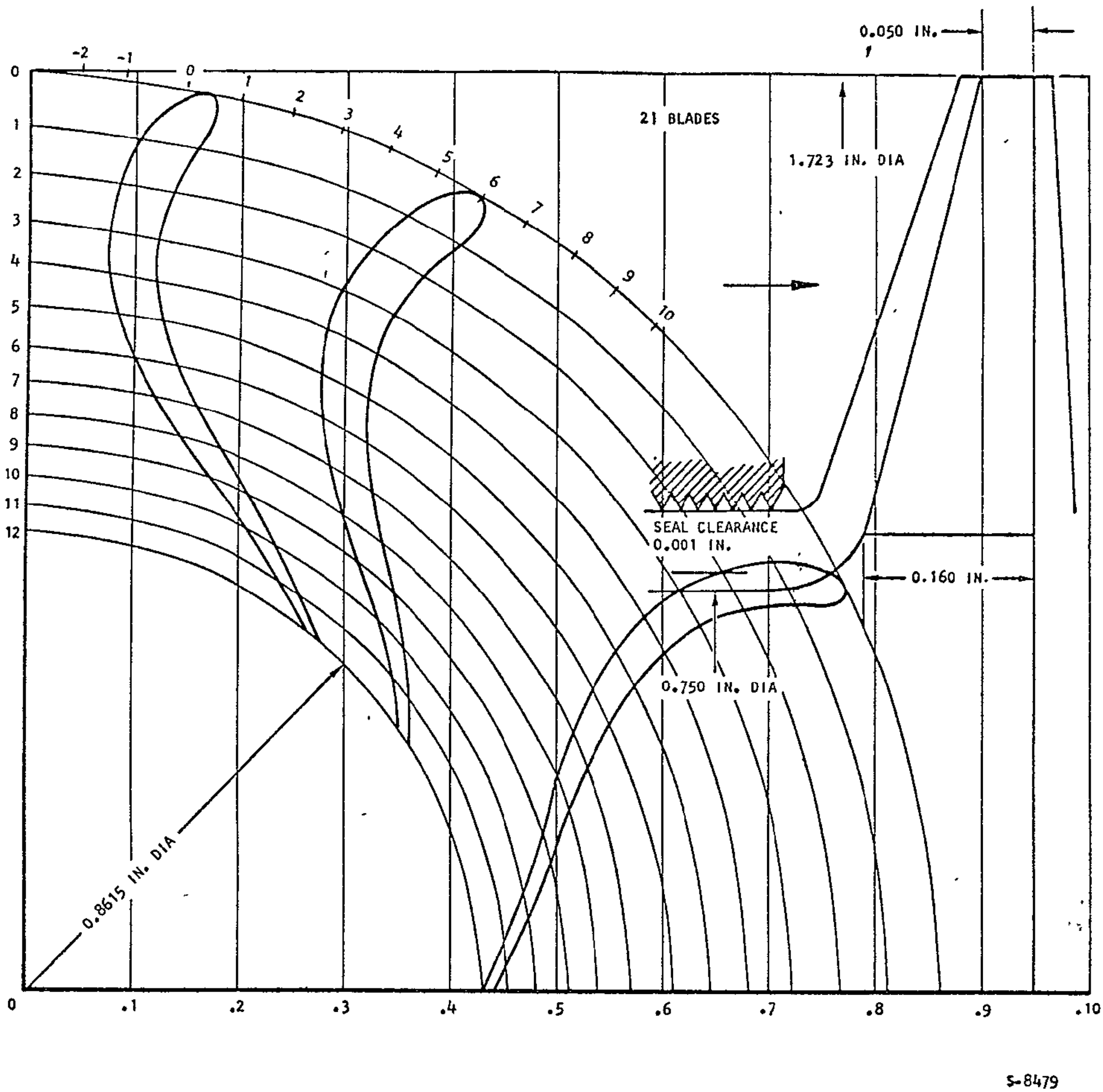


Figure 9-5. Turbine Wheel Layouts

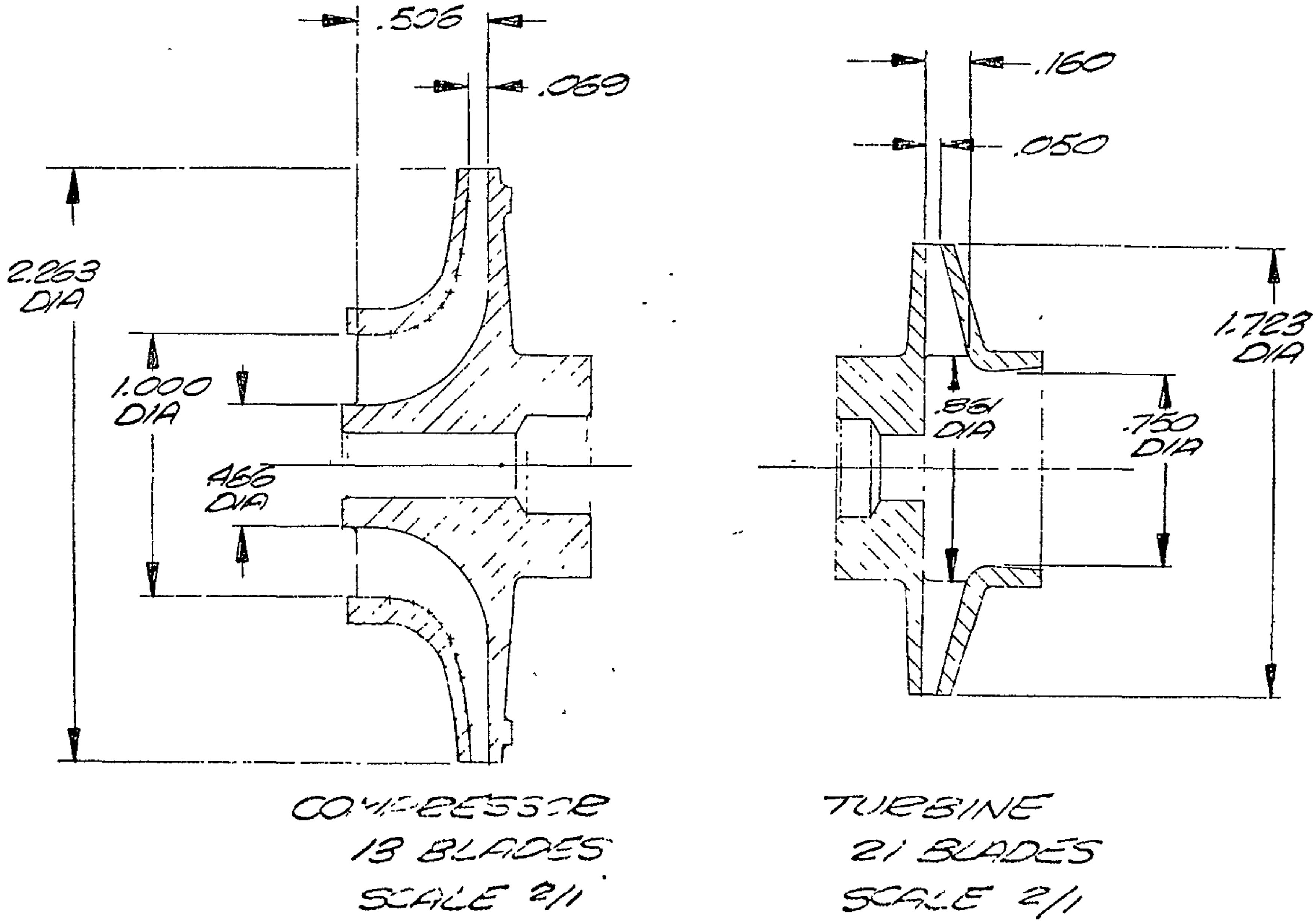


Figure 9-6. Compressor and Turbine Wheel Cross Sections

9.3 MULTIFAMILY RESIDENCE UNIT

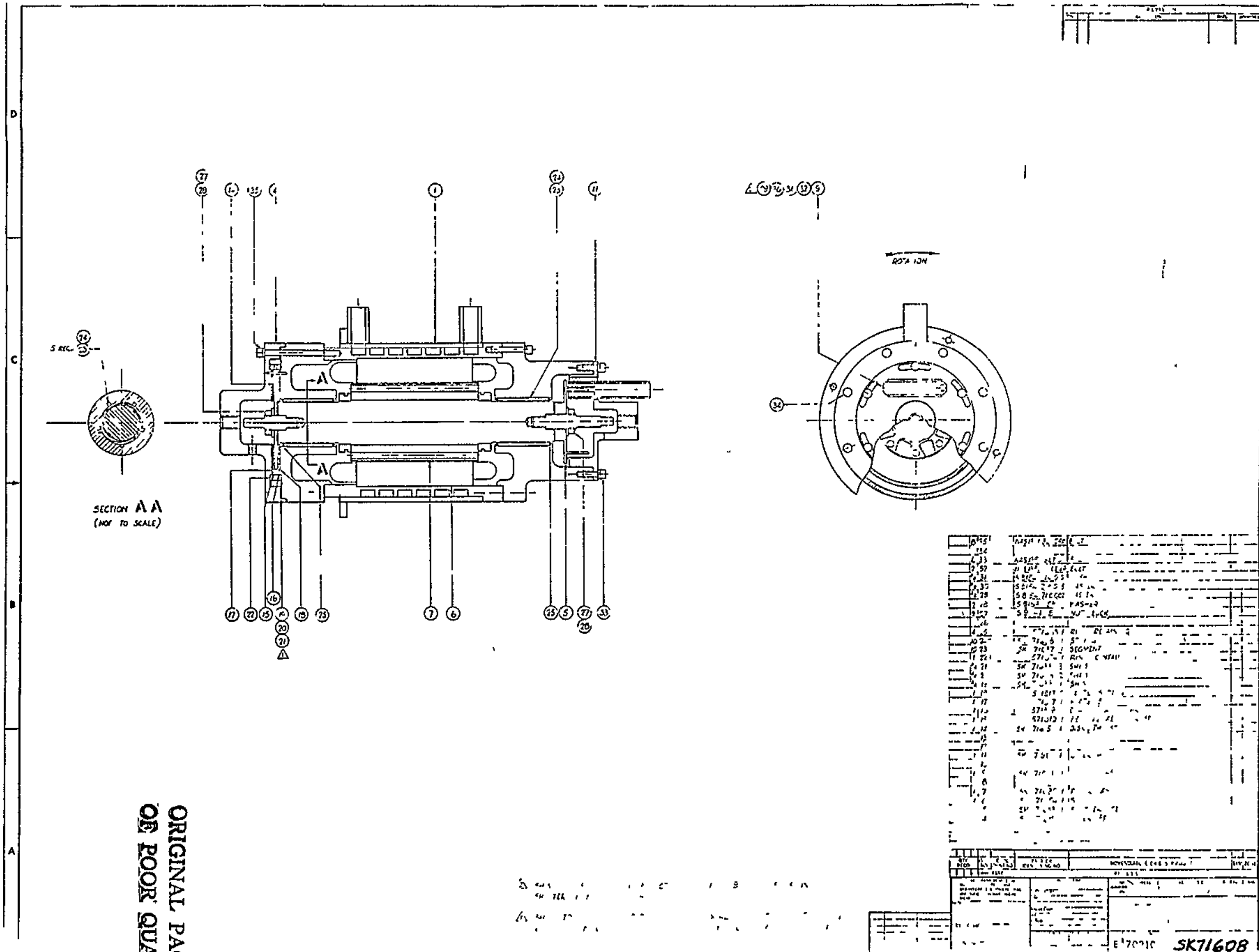
The effort on this machine has been concerned with the development of an R-11 test rig for evaluation of an existing PM motor supporting on foil bearings. This motor is similar in terms of size and speed to what will be used to drive the 25-ton compressor. The intent here is to use the same bearing configuration as for the test machine. The purpose of the test is to verify bearing performance in an R-11 environment. Originally these bearings were developed for the A-7 aircraft cooling turbine where air is the working fluid. The test motor and bearing assembly are shown in SK71608.

Because of the availability of the motor and bearings, it was felt that this test unit should be assembled and utilized to determine as early as possible actual motor characteristics, including starting characteristics, bearing/motor interactions, motor cooling data, speed and position sensing characteristics, and material compatibility with R-11.

All details for the assembly of this unit have been released, and it is in the process of fabrication. A detailed schedule for the design, fabrication, and testing of this unit is shown in Figure 9-7. The timing is such that the information obtained can be utilized in the design of all the units.

SK71126 shows the cross-section of the unit as originally proposed. Except for operating speed and position sensor design, the final units should be very similar to the proposed machine. Detailed analyses are in process, and data should be available at the time of the design review meeting.

Table 9-3 lists the requirements for the design of the 25-ton compressor, turbine, and motor. Data are given for both the heating and cooling modes of operation.



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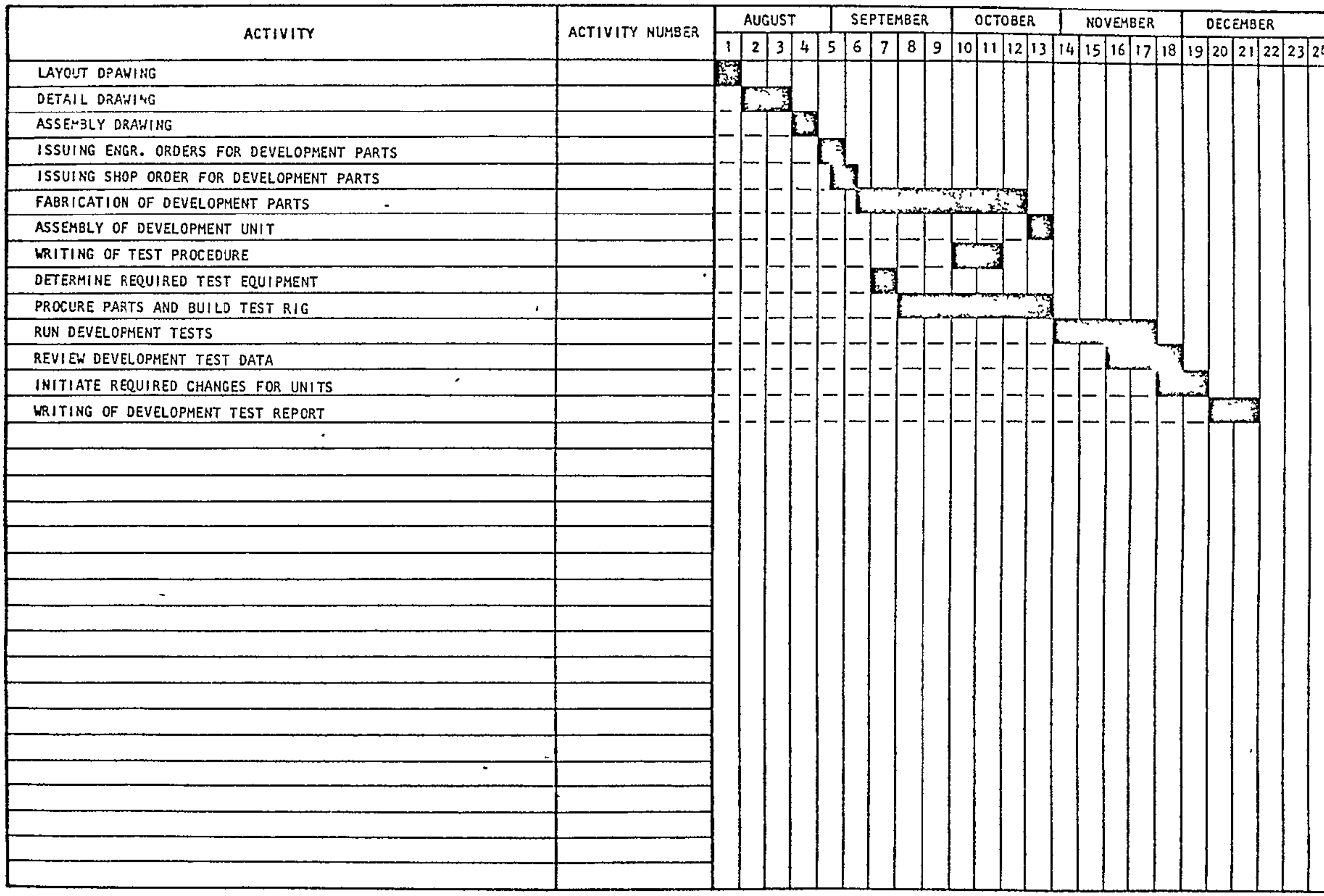

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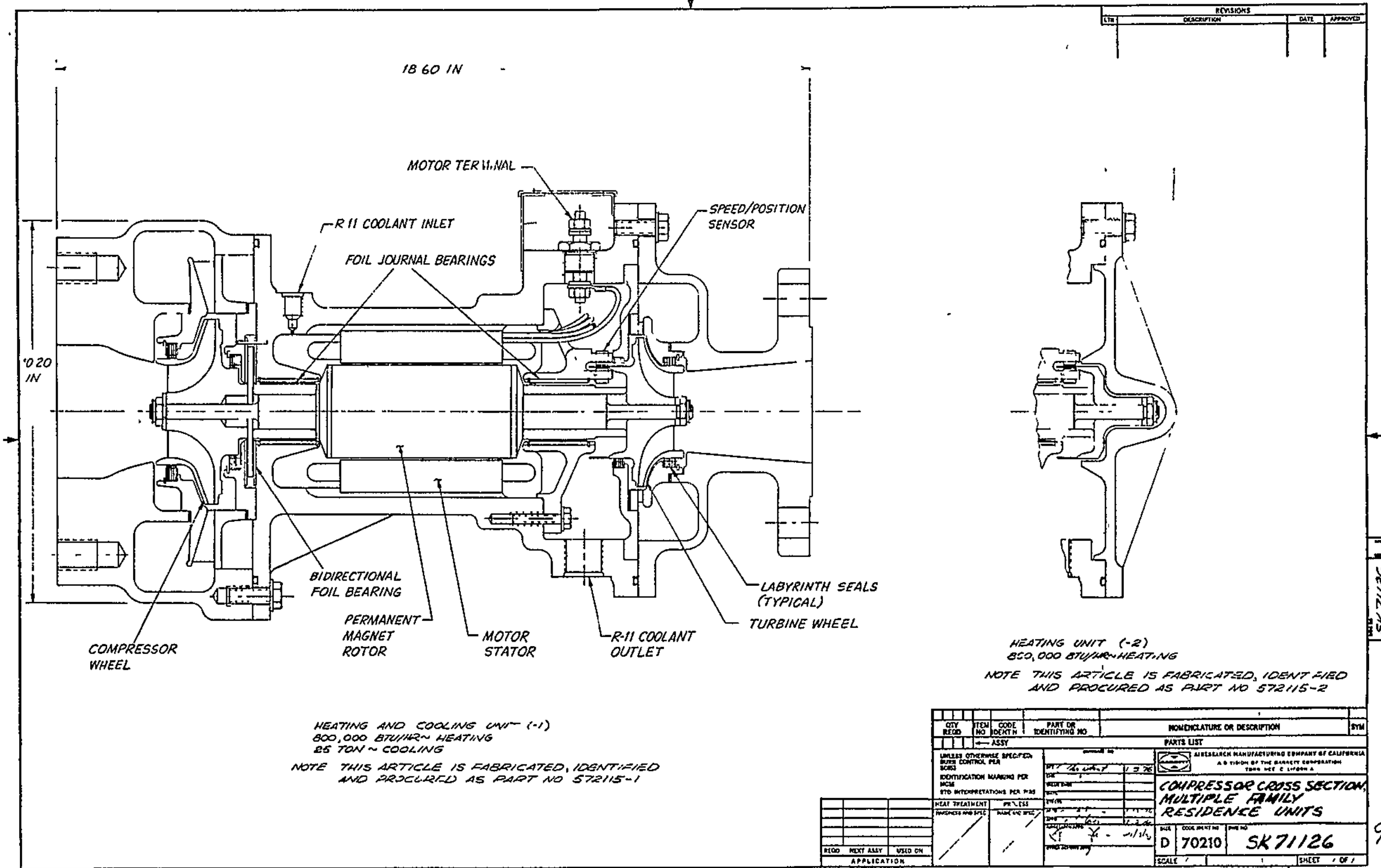
Figure 9-7. Bearing/Motor Test Unit Development Schedule


18 60 IN
'0 20 IN
MOTOR TERMINAL
R 11 COOLANT INLET
FOIL JOURNAL BEARINGS
SPEED/POSITION SENSOR
BIDIRECTIONAL FOIL BEARING
PERMANENT MAGNET ROTOR
MOTOR STATOR
R-11 COOLANT OUTLET
LABYRINTH SEALS (TYPICAL)
TURBINE WHEEL
COMPRESSOR WHEEL
HEATING UNIT (-2)
850,000 BTU/HR~HEATING
NOTE THIS ARTICLE IS FABRICATED, IDENTIFIED AND PROCURED AS PART NO 572115-2
HEATING AND COOLING UNIT (-1)
800,000 BTU/HR~ HEATING
25 TON ~ COOLING
NOTE THIS ARTICLE IS FABRICATED, IDENTIFIED AND PROCURED AS PART NO 572115-1
REVISIONS
LTR
DESCRIPTION
DATE
APPROVED
QTY REQD
ITEM NO
CODE IDENT N
PART OR IDENTIFYING NO
NOMENCLATURE OR DESCRIPTION
SYM
ASSY
PARTS LIST
UNLESS OTHERWISE SPECIFIED
IDENTIFICATION MARKING PER
HEAT TREATMENT
USED ON
APPLICATION
COMPRESSOR CROSS SECTION, MULTIPLE FAMILY RESIDENCE UNITS
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70210
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SCALE
SHEET 1 OF 1
SK 71126

9-18

TABLE 9-3

25-TON TURBOCOMPRESSOR

	Cooling Mode	Heating Mode
Compressor		
Fluid type	R-11	R-11
Inlet pressure, psia	7.50	8.27
Inlet temperature, °F	45	47.2
Inlet H, Btu/lb	98.01	98.28
Outlet pressure, psia	21.65	34.92
Outlet H, actual, Btu/lb	107.82	117.01
ΔH adiabatic, Btu/lb	8.02	11.05
ΔH actual, Btu/lb	9.81	18.73
Volume flow, inlet, cfs	5.67	9.03
Weight flow, lb/sec	1.1	1.93
Speed, rpm	30,000(constant)	41,000
Theoretical efficiency, percent	81.8	59
Power at wheel, kw	11.38	38.0
Turbine		
Inlet pressure, psia	85.3	NA
Inlet temperature, °F	190	
Inlet H, Btu/lb	115.4	
Outlet pressure, psia	21.65	
Outlet H, Btu/lb	106.14	
ΔH adiabatic, Btu/lb	10.94	
ΔH actual, Btu/lb	9.26	
Volume flow, inlet, cfs	2.38	
Weight flow, inlet, lb/sec	1.237	
Speed, rpm	30,000(constant)	
Theoretical efficiency, percent	84.7	
Power at wheel,kw	12.08	
Motor		
Maximum speed, rpm	30,000(constant)	41,000
Minimum speed, rpm	NA	18,500
Power at maximum speed, kw	14.5*	38.0**
Power at minimum speed, kw	14.5*	NA
Growth potential, power, percent	NA	20
Growth potential, speed, percent	10	10
Maximum overspeed, rpm	NA	45,100
Pressure, cavity, psia	9.5	9.5
Temperature, cavity, °F	100	150
Viscosity, μ, Reynolds	1.64×10^{-9}	1.75×10^{-9}
Motor, efficiency, percent	NA	92.91
Bearings/lubricant, types	Foil/R-11/vapor	Foil/R-11/vapor
Type motor	PM Samco	PM Samco
No-load power	TBD	NA
Maximum design speed, rpm	NA	50,000
General		
Design life, yr	15	15
No. of starts, lifetime	60,000	60,000
Service hours, lifetime	75,000	50,000
Type control	on/off	Modulating

* Power delivered to the compressor assuming 5 percent mechanical losses.

** Power delivered to the compressor assuming no mechanical losses.

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Preliminary analyses of the compressor and turbine were used to generate the performance maps of Figures 9-8, 9-9, and 9-10. These maps in turn were used to assess heat pump performance over the entire range of anticipated operating conditions.

9.4 COMMERCIAL UNIT

Compressor, turbine, and motor problem statements for this unit are listed in Table 9-4 for the heating and cooling modes of operation. Preliminary design analysis of the compressor and turbine yielded the maps shown in Figures 9-11, 9-12, and 9-13. These maps were used for heat pump performance evaluation.

9.5 MOTOR DESIGN

A series of motors has been designed to the new system requirements. The effects of varying air gap, rotor tip speed, and machine commutating reactance have been evaluated. A summary of the performance characteristics of the present motor is given in Table 9-5. Thermal analysis and rotor stress analysis are in process. Depending on the results of this work, modifications to the motor design may be made in addition to the critical speed analysis of the combined motor-turbocompressor rotating assembly.

The approach to rotor position sensing has been changed from that discussed in the proposal. For the 3-ton system, the electromagnetic size of the unit is so small that use of the proposed configuration would represent a significant penalty in shaft length and windage loss, thus resulting in a reduced efficiency. Work is progressing toward incorporating Hall generators for the position sensing. The Hall devices are compatible with the projected thermal environment, and their use will result in a minimum impact with respect to shaft length and windage loss.

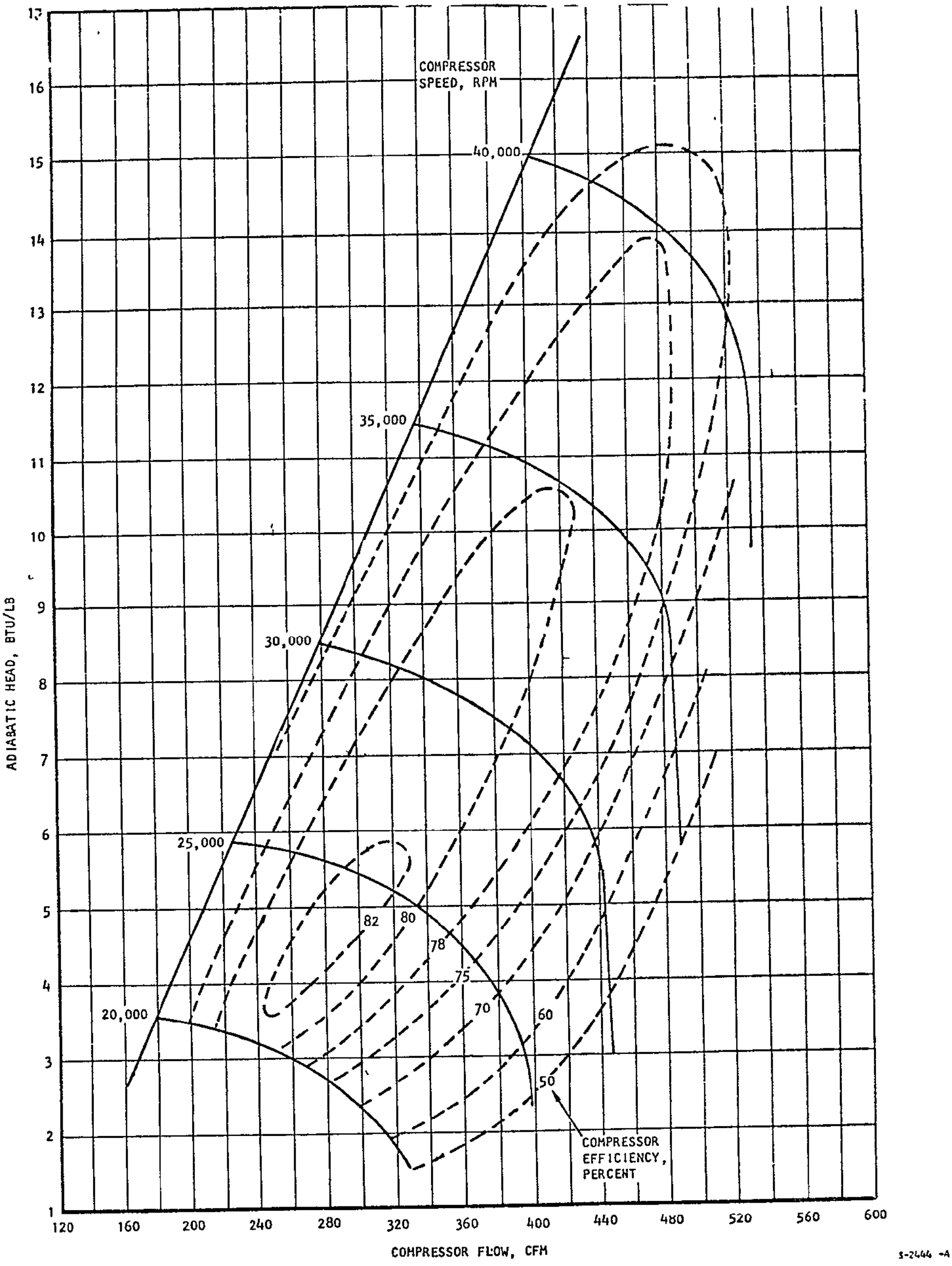

Figure 9-8. Compressor Map (25-Ton/800,000-Btu/hr Unit)

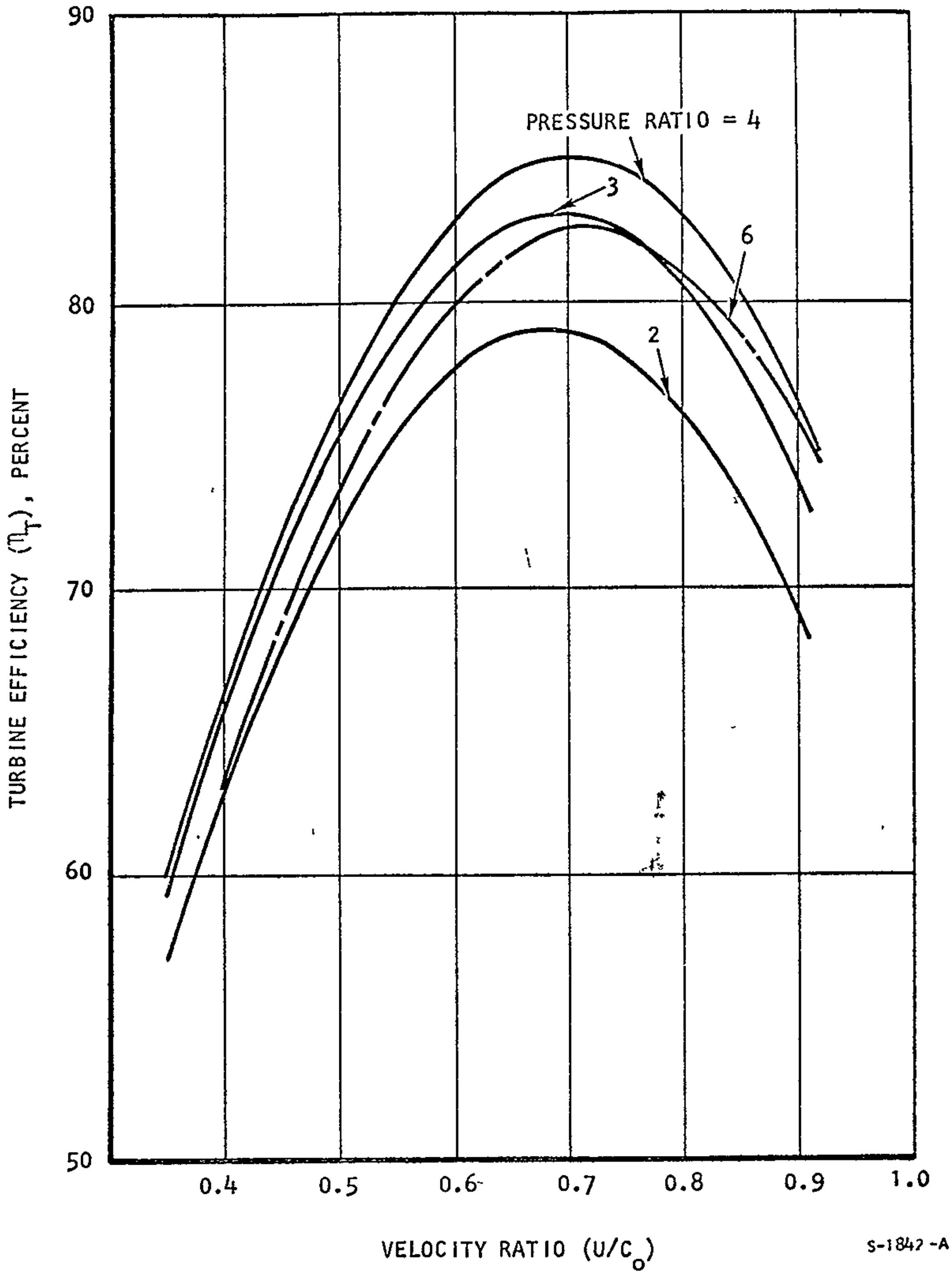

Figure 9-9. Turbine Efficiencies (25-Ton Unit)

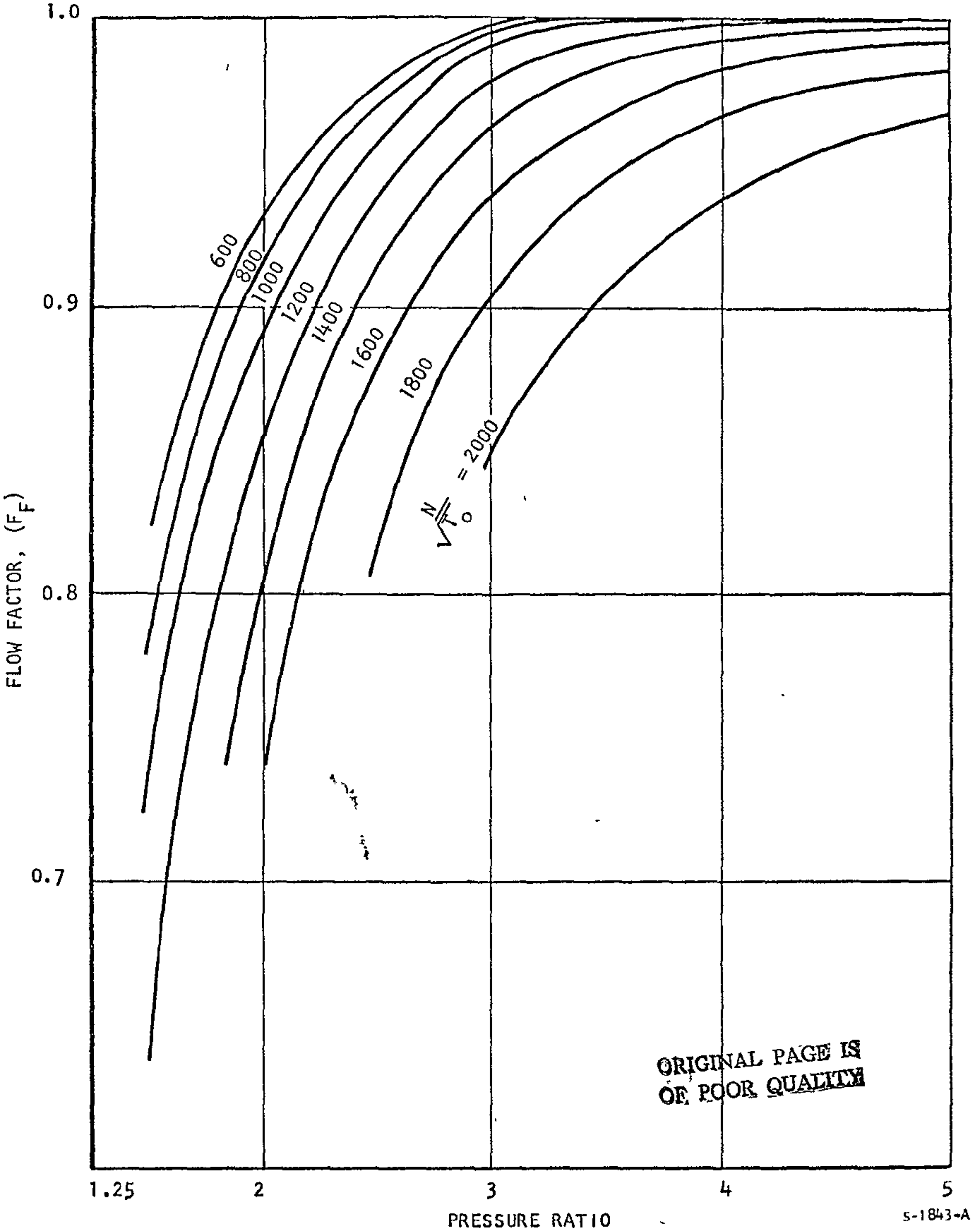

Figure 9-10. Turbine Flow Coefficients (25-Ton Unit)

TABLE 9-4

10-TON TURBOCOMPRESSOR

	Cooling Mode	Heating Mode
Compressor		
Fluid type	R-11	R-11
Inlet pressure, psia	8.04	8.74
Inlet temperature, °F	50.80	58.0
Inlet H, Btu/lb	98.57	99.7
Outlet pressure, psia	22.40	31.8
Outlet H, actual, Btu/lb	108.57	114.00
ΔH adiabatic, Btu/lb	7.8	10.20
ΔH actual, Btu/lb	10.0	15.0
Volume flow, inlet, cfs	2.19	2.93
Weight flow, lb/sec	0.453	0.657
Speed, rpm	45,430(constant)	57,440
Theoretical efficiency, percent	78	68
Power at wheel, kw	4.76	10.4
Turbine		
Inlet pressure, psia	85.6	NA
Inlet temperature, °F	190.0	
Inlet H, Btu/lb	115.3	
Outlet pressure, psia	22.0	
Outlet H, Btu/lb	104.7	
ΔH adiabatic, Btu/lb	10.7	
ΔH actual, Btu/lb	8.56	
Volume flow, inlet, cfs	2.82	
Weight flow, inlet, lb/sec	0.542	
Speed, rpm	45,430	
Theoretical efficiency, percent	0.80	
Power at wheel, kw	4.90	
Motor		
Maximum speed, rpm	47,500(constant)	57,440
Minimum speed, rpm	NA	27,500
Power at maximum speed, kw	5.5*	10.4**
Power at minimum speed, kw	5.5*	NA
Growth potential, power, percent	NA	20
Growth potential, speed, percent	10	10
Maximum overspeed, rpm	NA	63,140
Pressure, cavity, psia	9.6	9.6
Viscosity, μ, Reynolds	1.64×10^{-9}	1.75×10^{-9}
Temperature, cavity, °F	100	150
Motor efficiency, percent	NA	90
Bearings/lubricant, types	Foil/R-11/vapor	Foil/R-11/vapor
Cooling flow, lb/sec (approx)	0.0022	0.0060
Type motor	PM Samco	PM Samco
No-load power	TBD	NA
Maximum design speed, rpm	NA	69,500
General		
Design life, yr	20	20
No. of starts, lifetime	60,000	60,000
Service hours, lifetime	75,000	50,000
Type control	on/off	Modulating

* Power delivered to the compressor assuming 5 percent mechanical losses.

** Power delivered to the compressor assuming no mechanical losses.

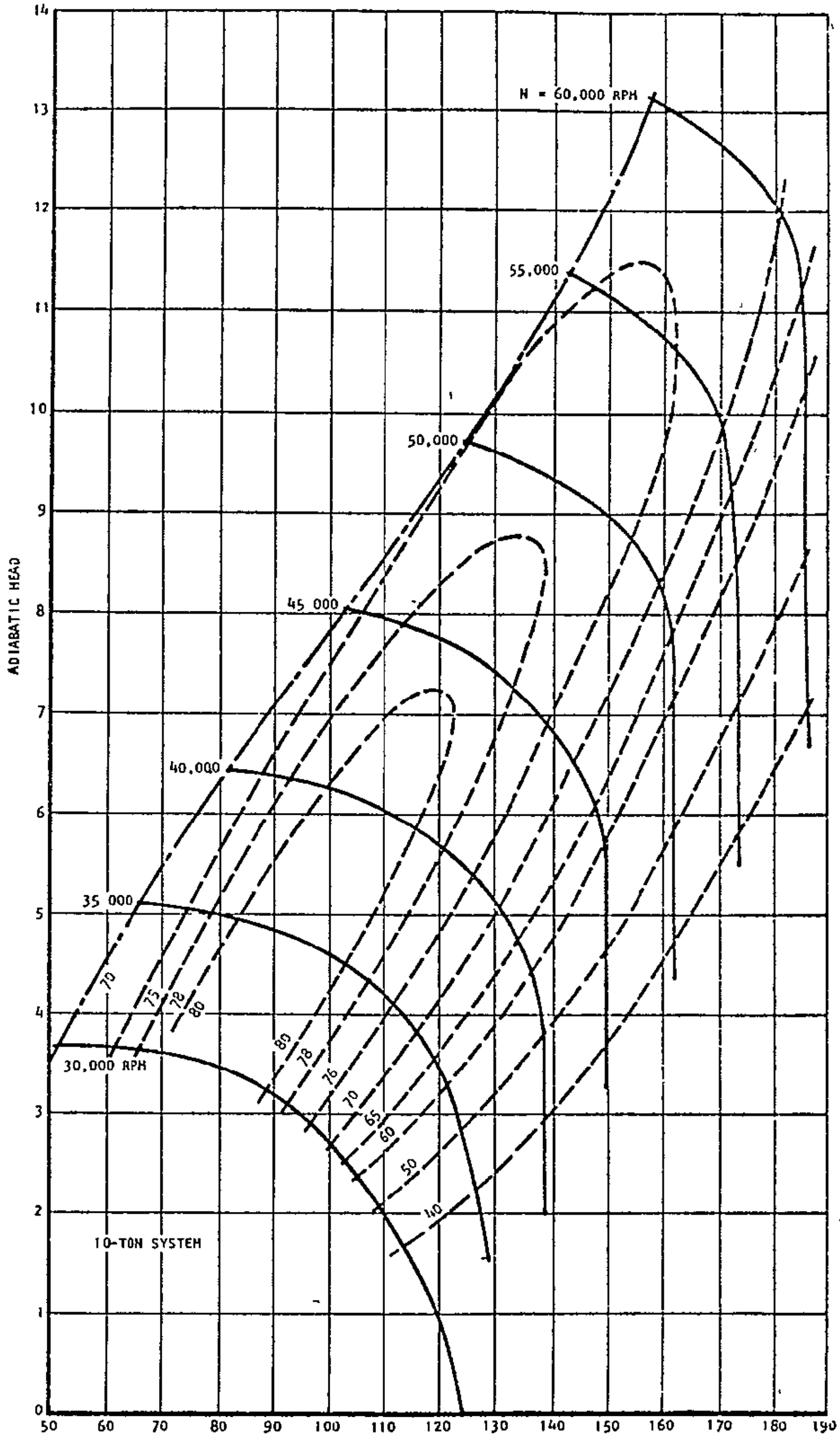

Figure 9-11. Compressor Map (10-Ton)

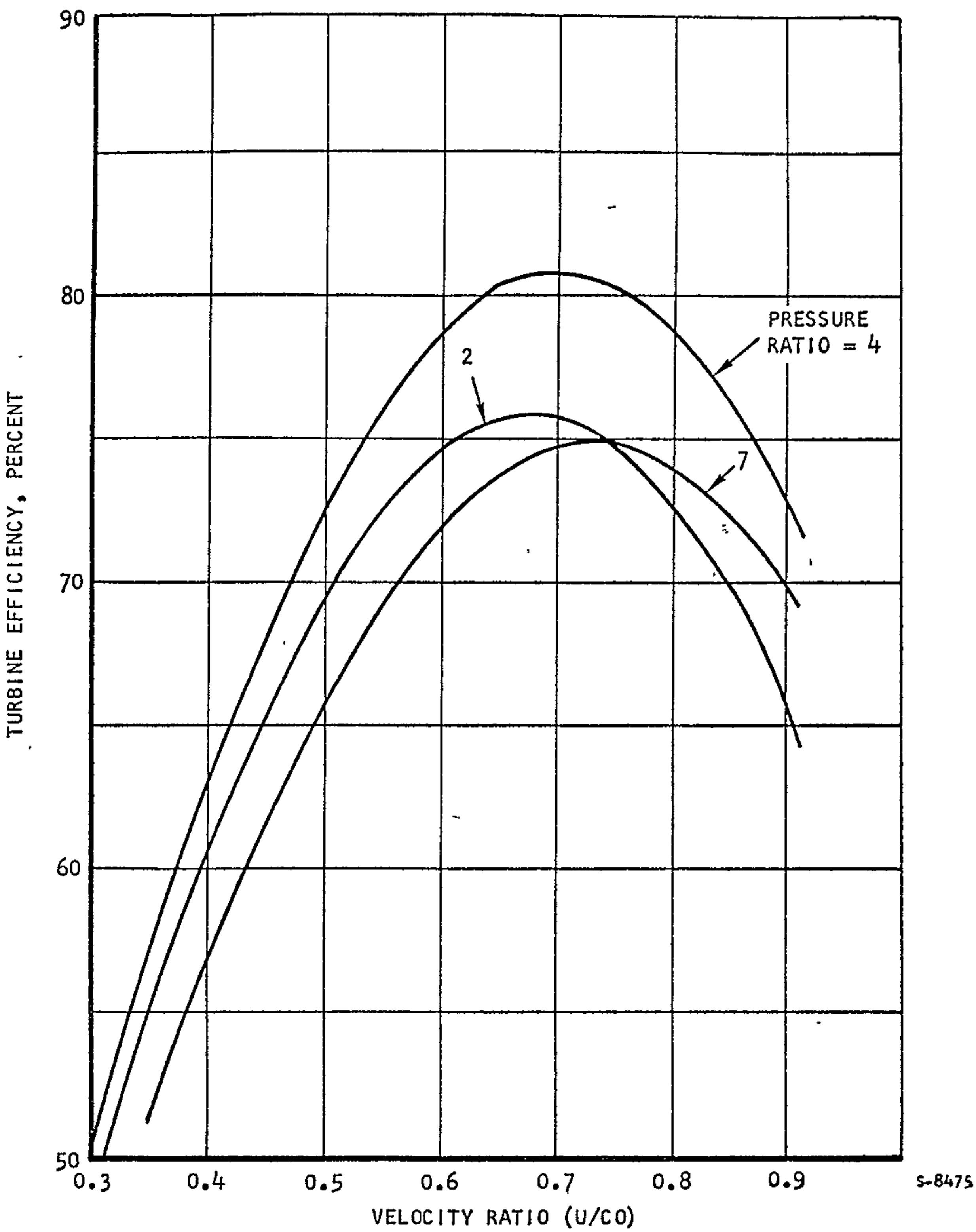

Figure 9-12. Turbine Efficiencies (10-Ton Unit)

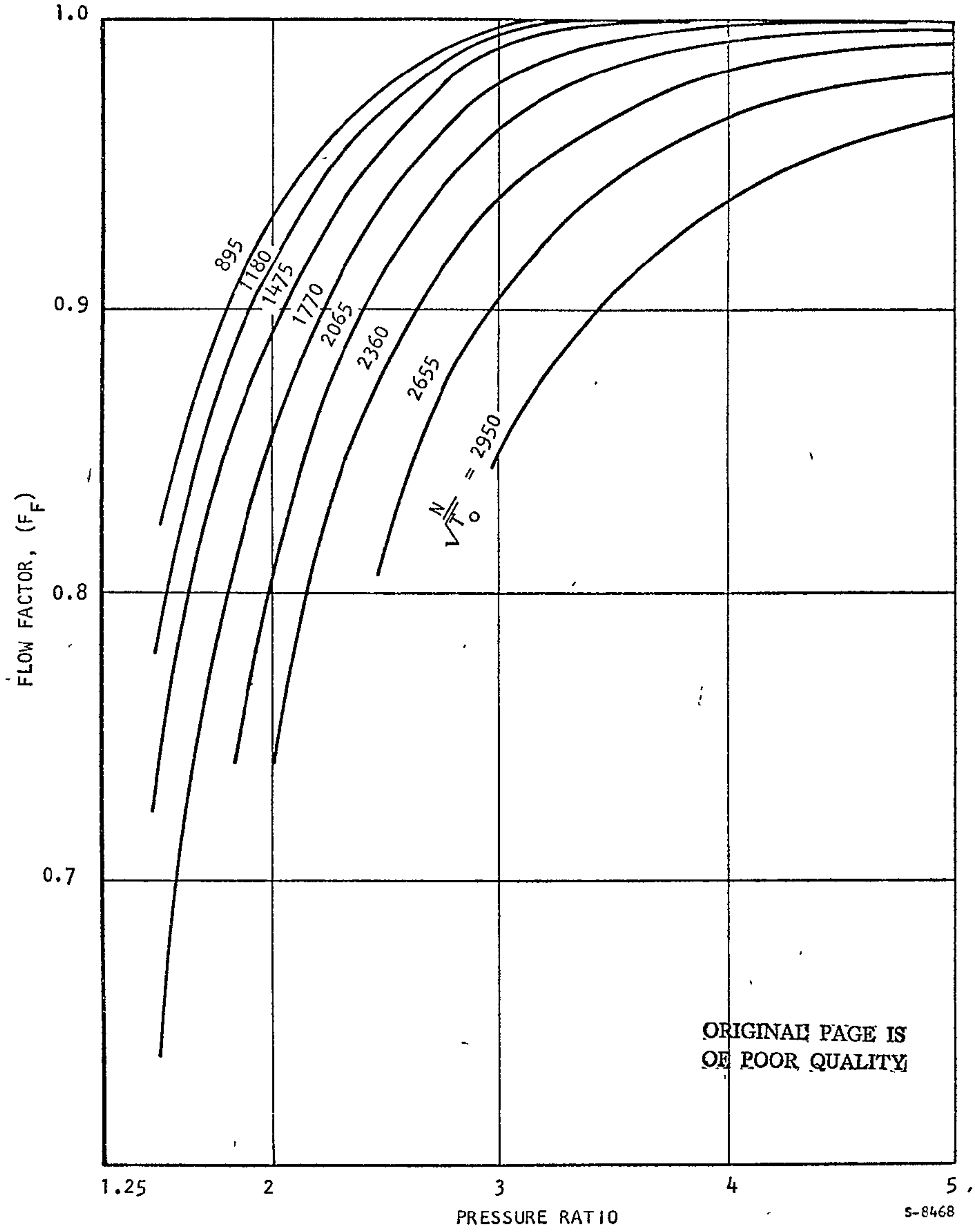

Figure 9-13. Turbine Flow Coefficients (10-Ton Unit)

TABLE 9-5

COMPRESSOR MOTOR CHARACTERISTICS 3-TON/60 KBTUH HEAT PUMP

Parameter	Units	Parameter	Units
RPM	86,200	Losses, w	
KW (shaft)	4.16	Windage	84
KVA (input)	4.80	Copper	58
PF (lagging)	0.95	Stray	10
Poles, no.	6.0	Pole head	17
Phases	3.0	Stator teeth	156
Frequency, Hz	4130	Stator core	71
Tip speed, ft/sec	350	Efficiency	0.913
Stator current density, amp/sq in	7619	Flux densities, kilo lines/sq in.	
Slots, no.	27	Stator core	40
Pole embrace	0.667	Stator teeth	88
Stator material	0.007 in. Trancor-T	Gap	39
Rotor iron	HP 9-4-20	Pole iron	90
Magnet material	samarium cobalt	Magnet	45
Dimensions, in.		Magneto motor force, amp turns	
Rotor diameter	0.972	Core	0.32
Rotor length	2.11	Teeth	4.0
Stack ID	0.9818	Gap	189
Stack OD	2.08	Pole	25.1
Stack length	1.92	Armature reaction	82
End turn extension	0.504	Total	300.42
Total stator length	2.93	Magnet energy product at rated load, mega gauss, oersted	11.1
Slot height	0.364	Short circuit current, per unit	3.63
Tooth width	0.052	No-load voltage, per unit	1.09
Slot Opening	0.020	Impedance data, per unit	
Weight, lb		Synchronous reactances	0.3
Stator	1.06	Commutation reactance	0.3
Rotor	0.370	Stator resistance (hot)	0.153
Total weight	1.43	Stator leakage reactance (XL)	0.11

10. MOTOR CONTROL

10.1 GENERAL

Design activities involved selection of an approach for (1) control mechanization of the three sizes of machines and (2) circuit design for the 3-ton/60 KBTUH and 25-ton/200 KBTUH machines.

10.2 CONCEPT SELECTION

Two system concepts have been studied in detail: (1) SCR system, and (2) transistor system.

10.2.1 SCR System Characteristics

The SCR system consists of a phase delay rectifier that controls the current to an inverter. This current in turn provides commutated current to the brushless dc machine (See block diagram of Figure 10-1).

The major features of the SCR system are as follows:

- Motor current is conditioned twice--by the PDR and by the inverter
- Motor current is controlled by the PDR
- SCR's are used as switches
- Rotor position sensor is required to start the inverter, but would self commutate once the motor has reached 10 percent speed
- It may be advantageous to change from rotor position sensor to self-commutation for switching of inverter switches to increase motor torque at certain speeds
- SCR's require complex drive circuitry

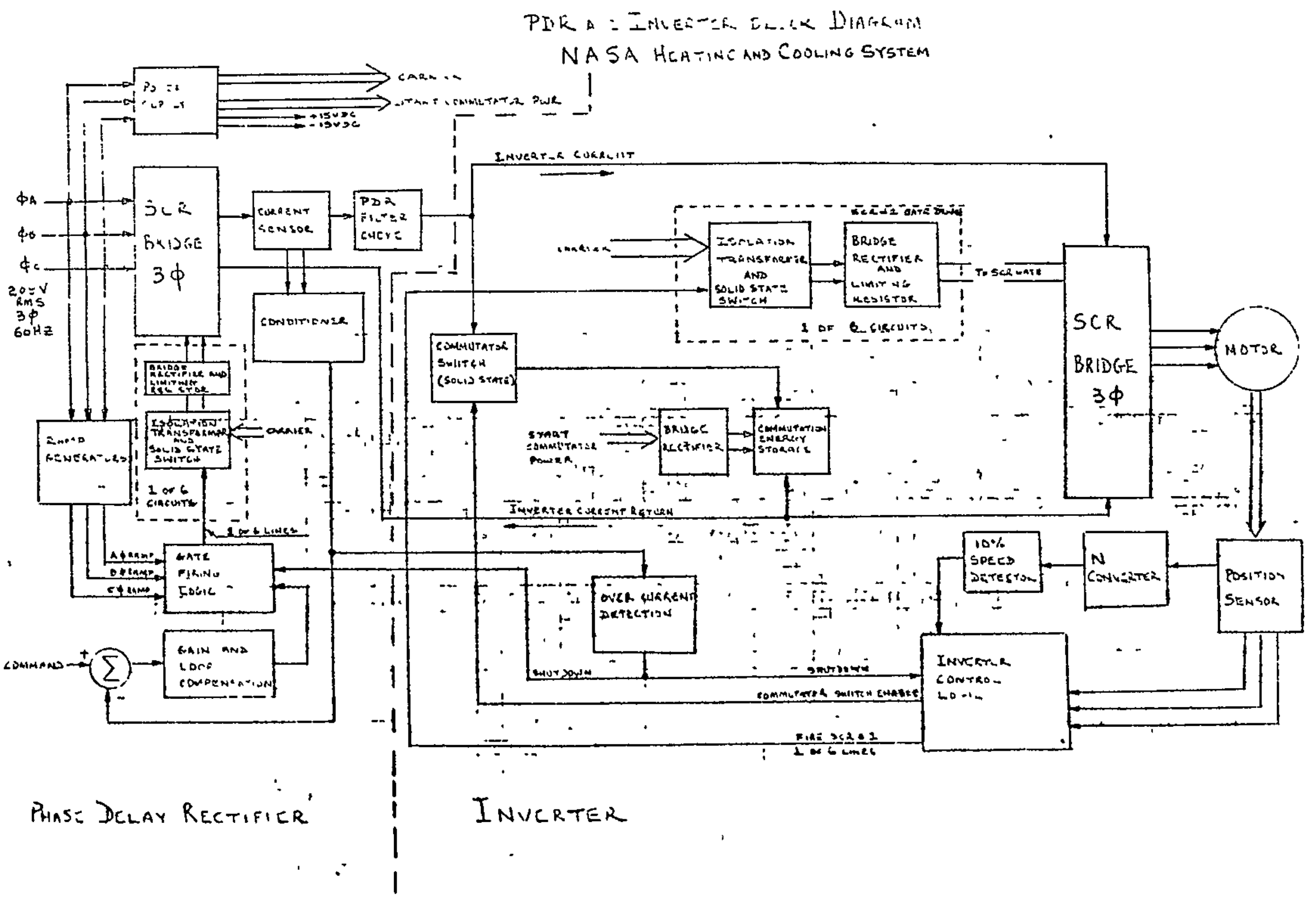

Figure 10-1. PDR and Inverter Block Diagram

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o High power SCR's are immediately available

o Rotor position sensor may not have to function at high speeds

10.2.2 Transistor System Characteristics

The transistor system consists of a rectifier, which produces dc, followed by a transistor inverter. The inverter provides commutation for the motor and also controls the current by pulse width modulating one of the pair of inverter switches that is on at any one time (see block diagram of Figure 10-2).

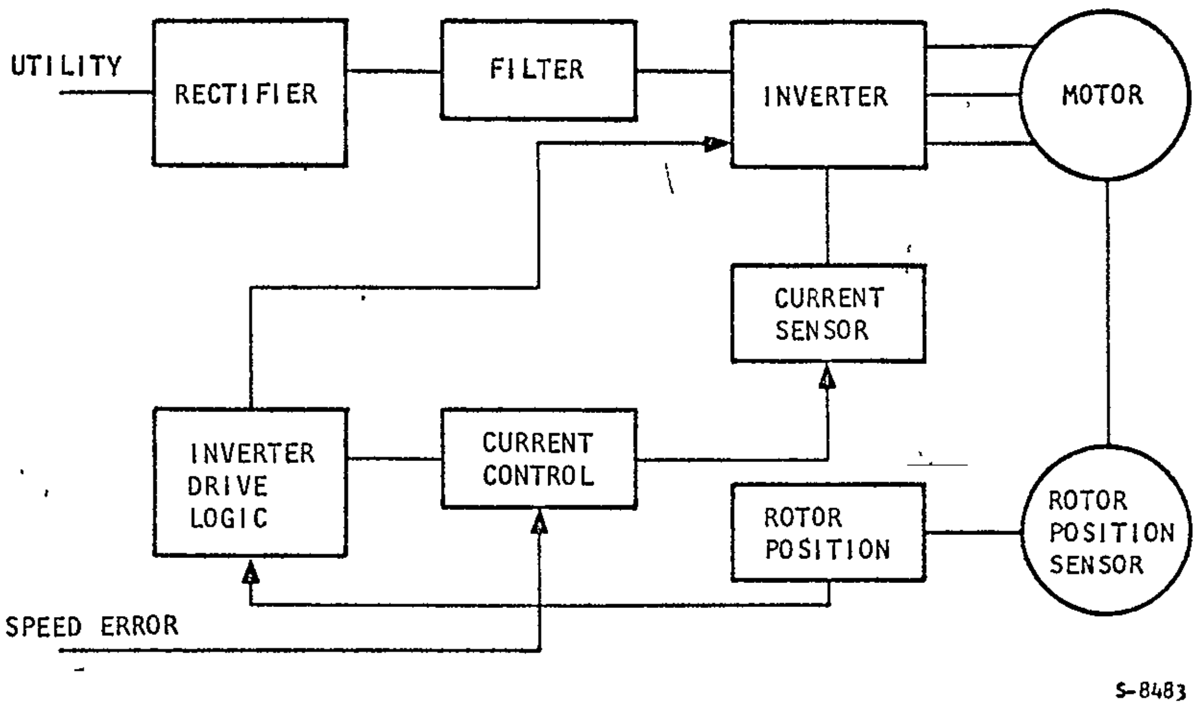

Figure 10-2. Transistor Control Block Diagram

Salient features of the transistor system are listed below.

- Motor current is conditioned twice--rectifier and inverter
- Motor current is controlled by pulse width modulation of the inverter
- Transistors are used for switches

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- Rotor position sensor is required for starting
- Motor back EMF could be used for position sensor once motor is up to speed
- There are possible advantages to using back EMF for motor torque control
- Transistors require simpler drive circuitry
- High power transistors are not available in commercial quantities today at acceptable prices; they could be available in 1986 if there is a large enough market
- Rotor position sensor may have to work at high speed

10.2.3 Concept Selection

The power rating of the inverters is the factor that controls the choice of mechanizations. It has been decided to build the low-power inverter (5-ton/60-KBTUH machine) with the transistor scheme and the high-power inverter with the SCR scheme. A decision on the medium-power inverter will be made when more data are available on projected power transistor technology advancement.

10.3 SCR CONTROL CIRCUIT DESCRIPTION (SEE FIGURE 10-1)

The direct current supply for the motors is generated by a phase delay rectifier (PDR). The current supplied by the PDR is sensed by a Hall device and conditioned. The conditioned current signal is compared to a current waveform, and a current error signal is generated. The error signal is compared to a cosine ramp waveform, and the output of this comparator is used to turn on solid-state switches that drive the SCR gates.

A switching power supply is used to generate dc power for the control circuitry. This power supply also generates two square wave voltages, one of which is isolated from the supply and used to supply power to the start commutator, the other to drive a transformer that supplies the gate drive

for the SCR's. This transformer is used to translate the voltage referenced to power supply ground to the SCR gate potential.

10.3.1 Ramp Generators

The dc output voltage of a phase delay rectifier is proportional to the cosine of the firing angle. Thus, if a cosine ramp is used in the generation of the firing angle, the output voltage becomes a linear function of the firing angle.

For a 3-phase PDR this cosine ramp is a cosine waveform whose maximum occurs at the intersection of the positive half-cycles of two of the phases. The ramp can then be used to generate the firing pulse for the + SCR connected to the more positive of the two phases.

10.3.2 Start Commutator

Below 10 percent speed, the back EMF is not adequate for SCR commutation. An auxiliary commutation scheme is therefore needed to provide the required voltage for commutation.

This voltage is generated by charging a large capacitor, which is isolated by a solid-state switch from the inverter supply current. When commutation is required, the solid-state switch is turned on and the capacitor, which has previously been charged to a polarity opposite that required to drive the motor, is connected across the motor supply. The charge on this capacitor is adequate to enable it to absorb the PDR output current and reverse the polarity of the inverter supply, thereby causing SCR commutation.

10.3.3 Power Supply

The power supply employs conventional switching regulator circuitry. The input ac voltage is rectified and filtered by an lc filter stage. The resulting dc voltage is series regulated down to +14 vdc. This voltage is

used to provide startup power for the power supply circuitry. The line filter output voltage is pulse width modulated at a fixed frequency so that the product of the input voltage and the duty cycle is a constant. The output voltage of the switching regulator is again rectified and filtered, producing a dc voltage.

This dc voltage is applied to an inverter stage that is driven at the same frequency as the switching regulator.

The output of the inverter stage is then rectified and filtered to produce ± 15 vdc for powering the control circuitry. The +15 vdc is fed back to take over the load driven by the series regulator. Thus the dissipation of the power supply is rendered independent of the line voltage level, and good efficiency is assured at all input voltage levels required.

The inverter transistors are isolated by diodes and used to generate the cosine waveform that is needed to drive the SCR gate transformer.

10.4 CIRCUIT DESIGN

Figures 10-3 through 10-10 give details of completed circuit designs for the SCR control (600 KBTUH heat pump compressor motor). The following circuits are shown:

(a) Power Supply, Carrier Generator, Start Commutator (Figure 10-3)

(b) 3- Phase Delay Rectifier (Figure 10-4)

(c) 3- PDR SCR Gate Drive (Figure 10-5)

(d) 3- PDR Ramp Generator (Figure 10-6)

(e) 3- PDR Ramp Generator-Alternate (Figure 10-7)

(f) Integrating Network for 3- PDR (Figure 10-8)

(g) 3- PDR SCR Gate Logic (Figure 10-9)

(h) Inverter (Figure 10-10)

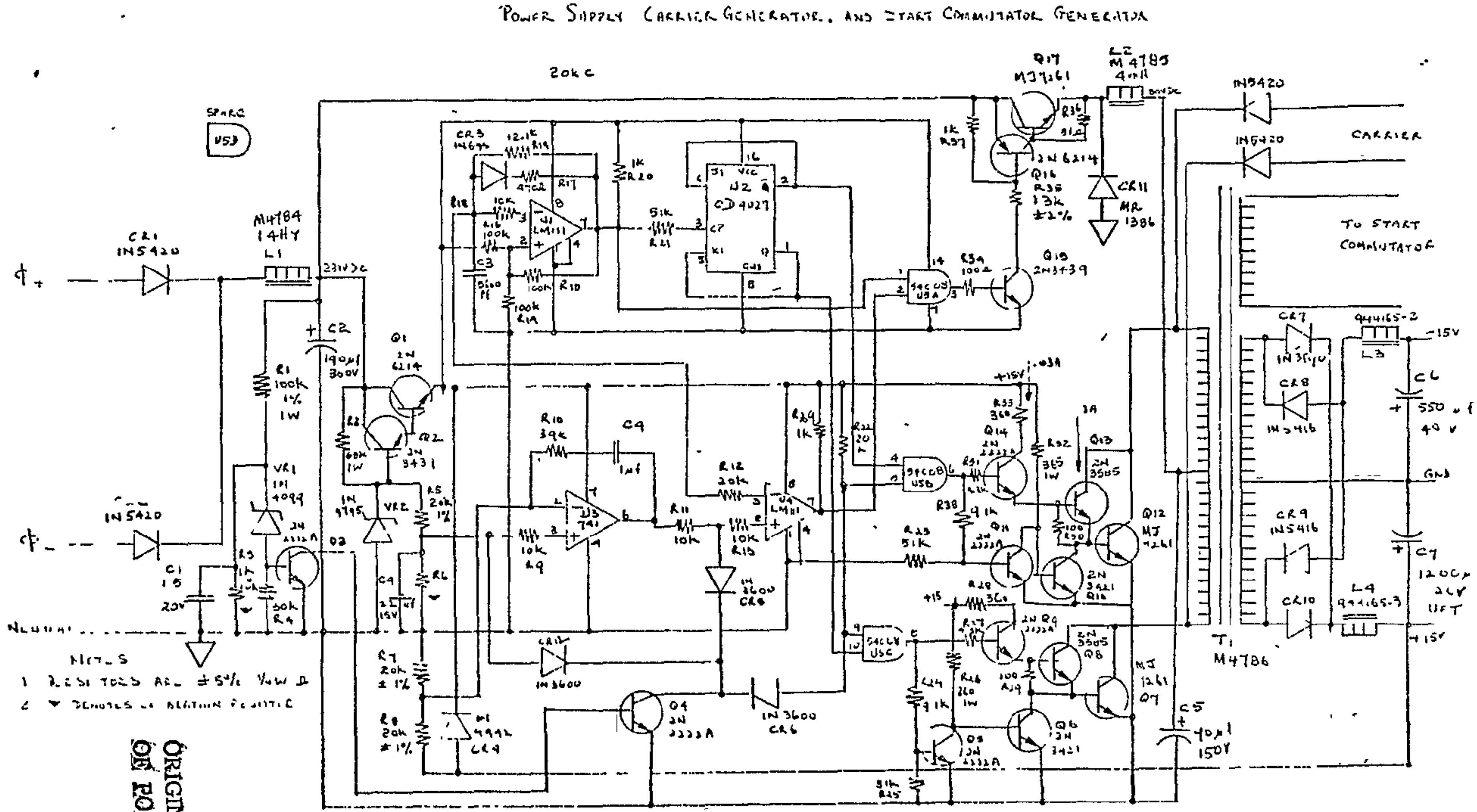

Figure 10-3. Power Supply, Carrier Generator, Start Commutator

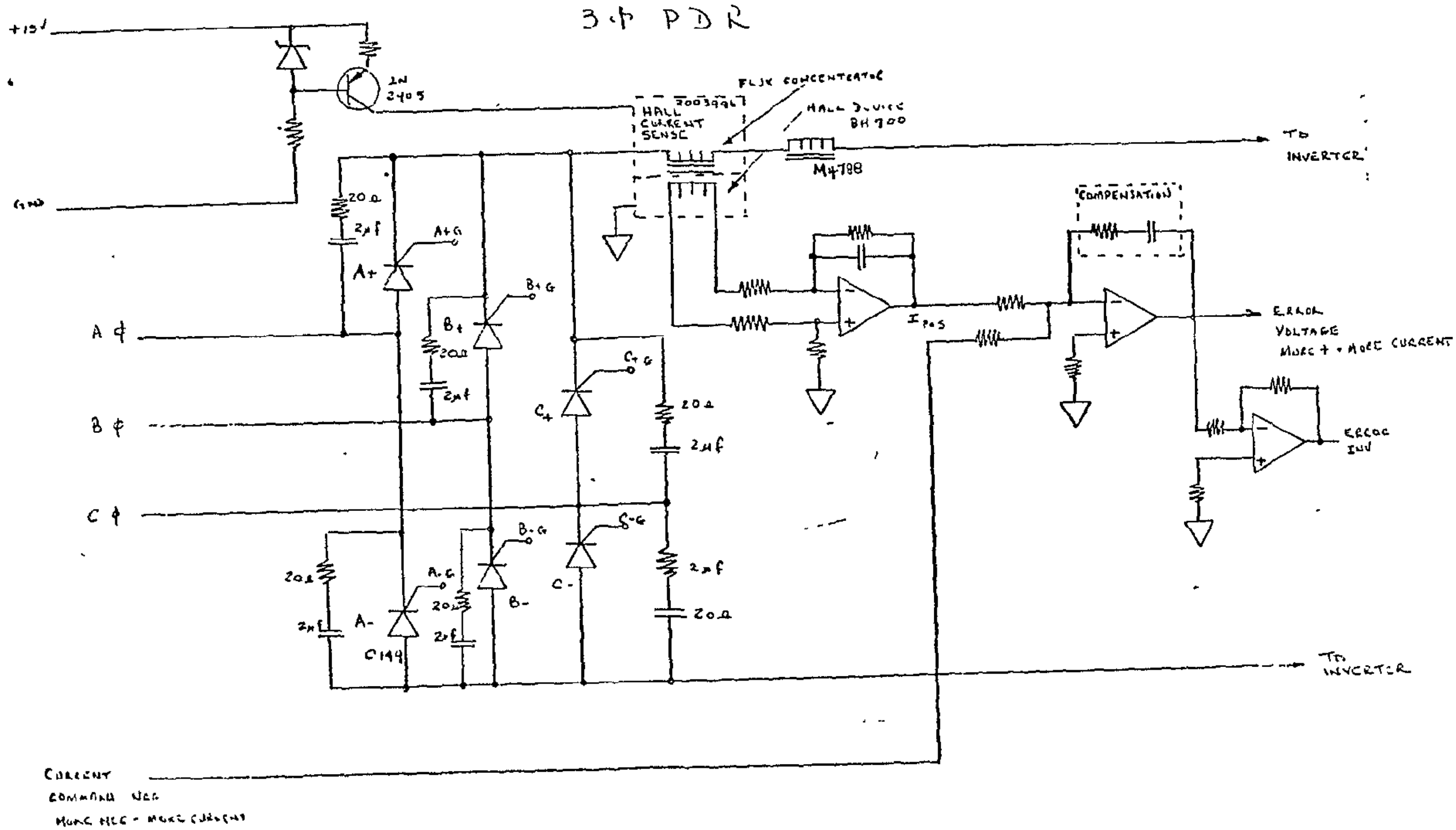

Figure 10-4. 3-Φ Phase Delay Rectifier

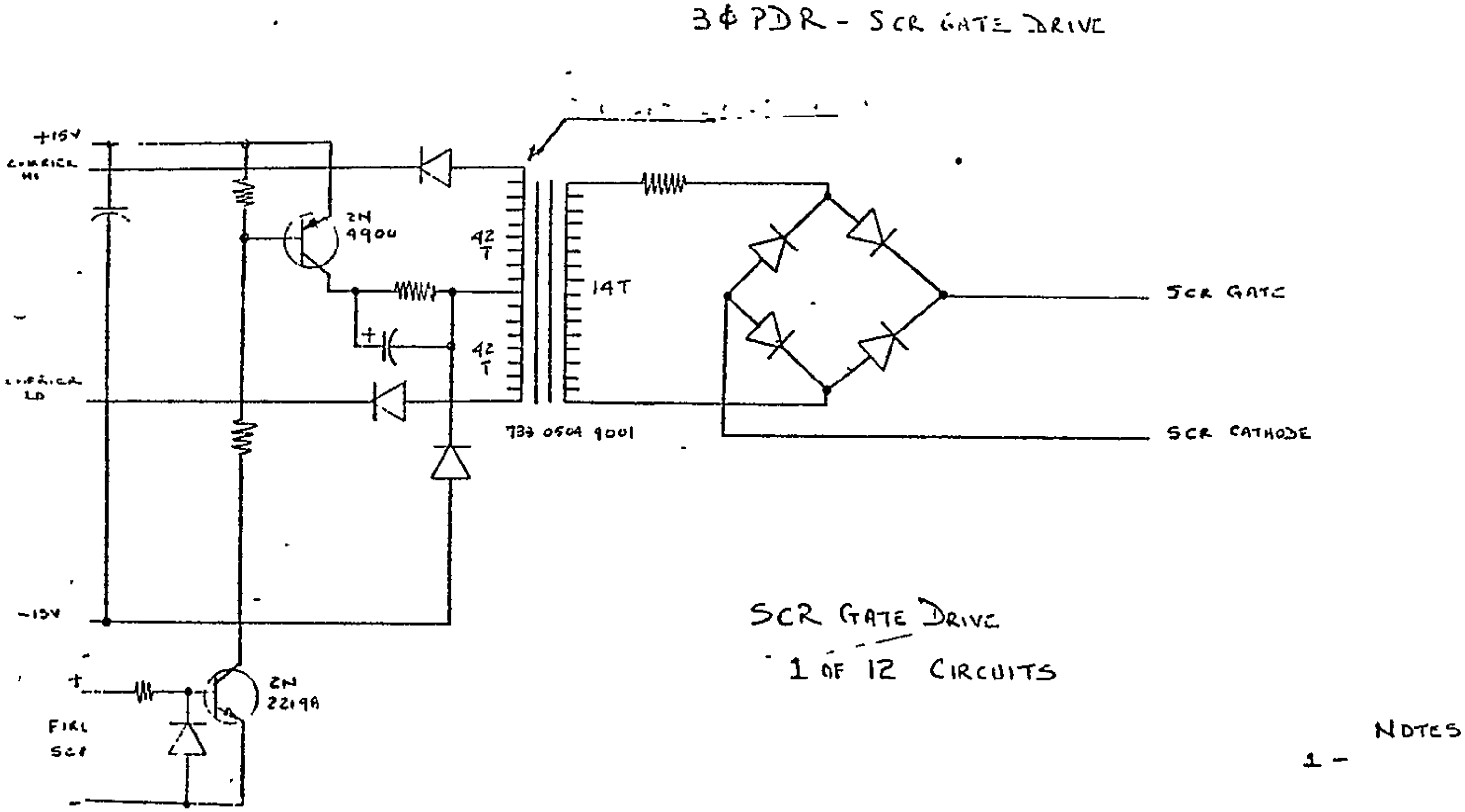

Figure 10-5. 3-Φ PDR SCR Gate Drive

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3Φ Phase Delay Rectifier, Ramp Generator

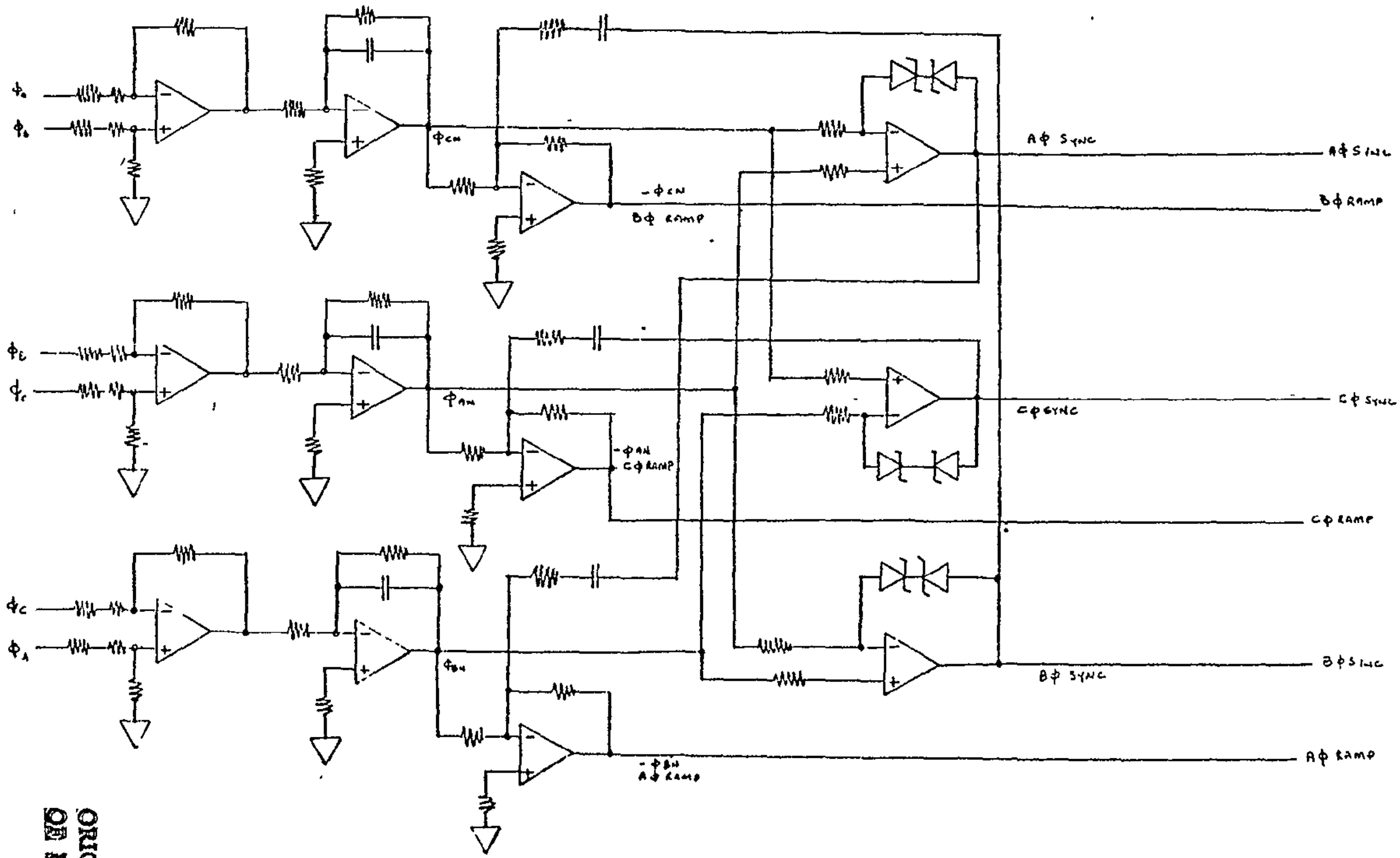

Figure 10-6. PDR Ramp Generator

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3φ PDR RAMP GENERATOR (ALTERNATE APPROACH)

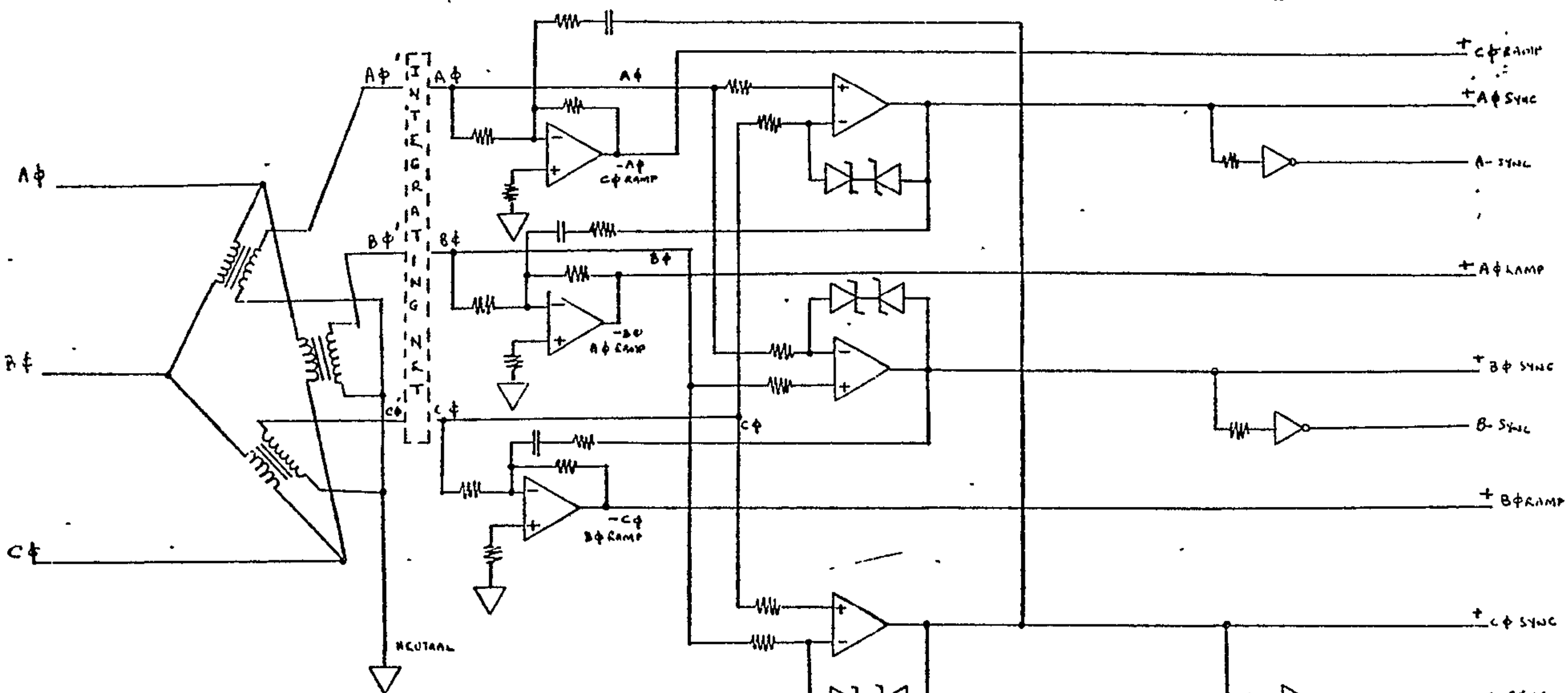

Figure 10-7. PDR Ramp Generator-Alternate

INTEGRATING NETWORK FOR 3Φ PDR.

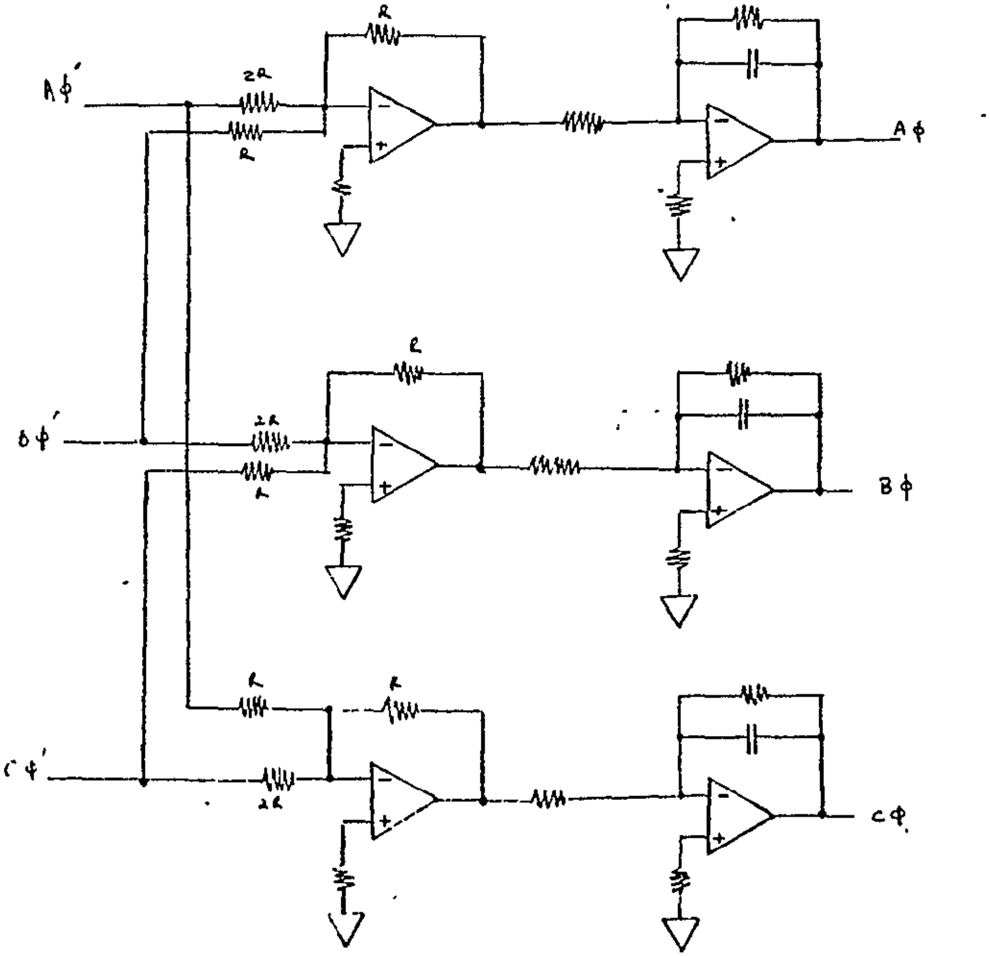

Figure 10-8. Integrating Network for 3-Ø PDR

3Φ PDR - SCR GATE LOGIC

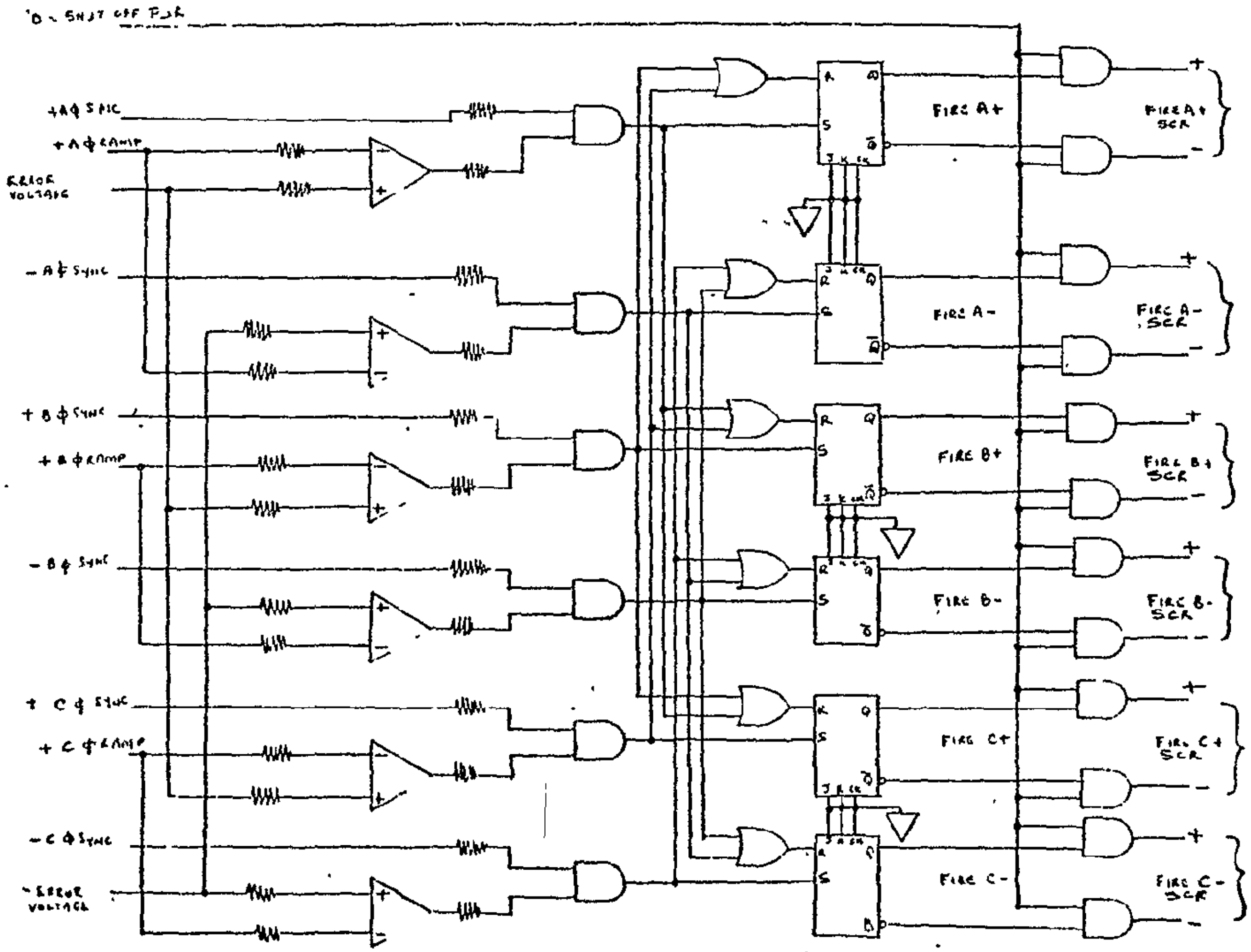

Figure 10-9. 3-Ø PDR SCR Gate Logic

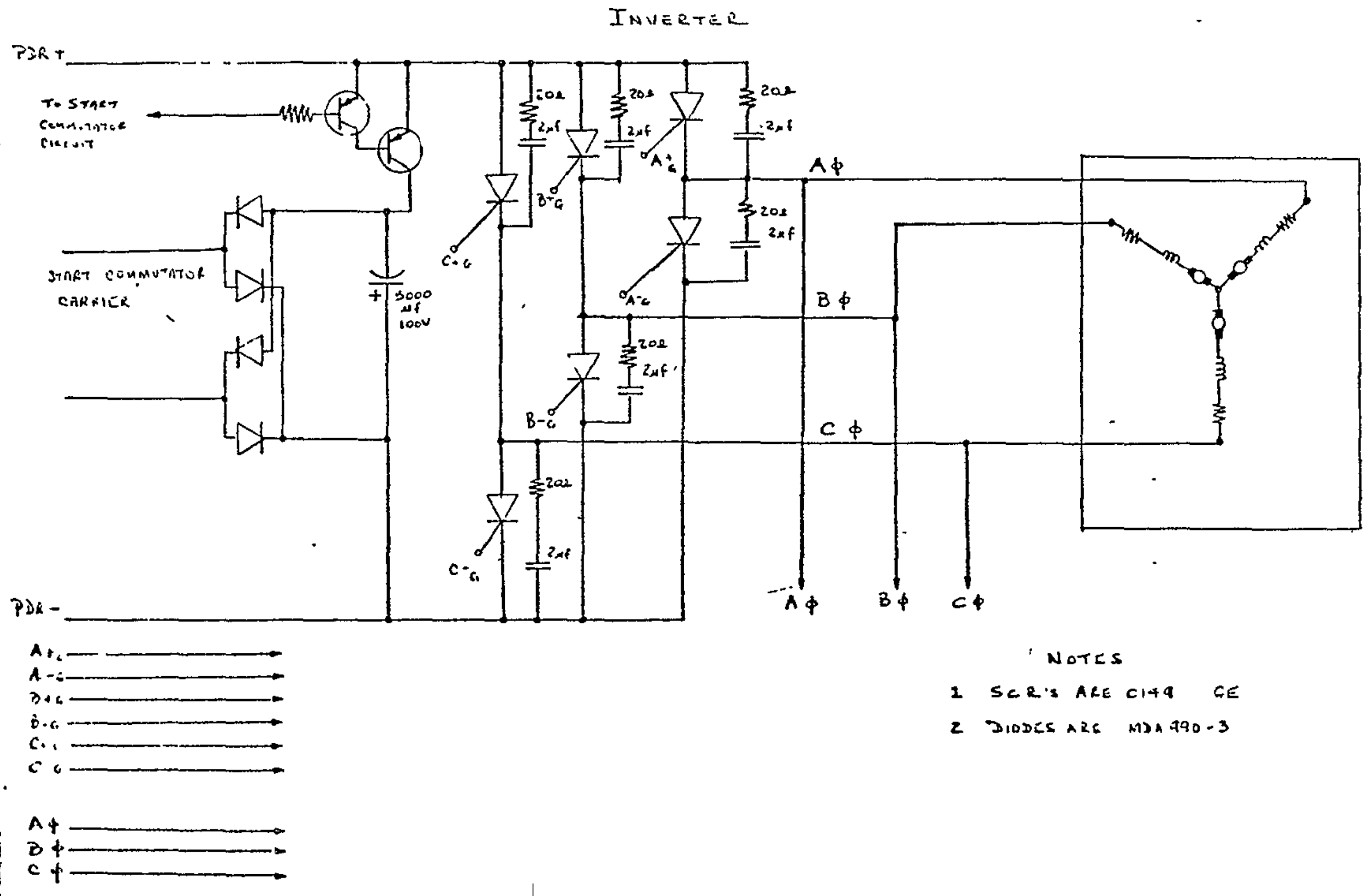

Figure 10-10. Inverter

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11. HEAT PUMP HEAT EXCHANGERS

11.1 OVERALL DESIGN APPROACH

The heat exchangers (evaporators, condensers, boilers, and heating/cooling air coils) for the proposed heating/cooling systems are sized to meet the specified requirements at minimal cost. Standard Dunham-Bush air conditioning and heat pump products, which are built primarily for refrigerants R-12, R-22, and R-508, are used in this system with minor modifications to accommodate increased water flow rates and the lower vapor pressure R-11 refrigerant. Volume production tooling is available for the heat exchangers selected even though they are not standard catalog sizes. They will be fabricated as any other Dunham-Bush heat exchanger production run.

11.1.1 Design Procedure

The heat exchangers described in this section were designed according to the following procedure:

(1) Problem statements were generated as a result of computerized system optimization studies.

(2) Basic heat transfer data available for the Dunham-Bush standard heat transfer surfaces were reduced in a form suitable for use by the AiResearch heat exchanger design computer programs (see Appendix B).

(3) The computer programs were exercised to generate a number of heat exchanger configurations and sizes corresponding to various basic heat transfer surfaces. All these candidate heat exchanger designs met the performance requirements developed in (1) above.

(4) The candidate designs were reviewed by Dunham-Bush for selection on the basis of cost, packaging constraints, and suitability for the kind of service considered.

(5) As a result of the Dunham-Bush investigations, iterations affecting the system performance were made to arrive at an overall optimum solution.

11.1.2 <u>Heat Exchanger Design Approach</u>

The system heat exchangers use three types of basic Dunham-Bush heat transfer surfaces. These are described in the following paragraphs.

11.1.2.1 <u>Finned-Tube Coil Heat Exchangers</u>

These heat exchangers transfer heat between a liquid (or refrigerant) circulated within the tubes and air circulated outside the tubes over the extended surfaces. The extended surfaces (fins) are always exposed to the gas side to compensate for the heat transfer coefficient between the gas and metal, which is relatively low compared with that between the liquid and metal. This basic heat transfer surface is presently used in many different Dunham-Bush air conditioning and heat pump products.

A typical fin selection is depicted in Figure 11-1. The wavy aluminum fins are formed by high precision dies that draw fin collars in the fin stock. The aluminum fin stocks that run vertically are tightly bonded to the horizontal copper tubes by mechanical expansion of the tubes. The fin collars not only provide accurate control of the fin spacing, but completely cover the tube for greatest heat transfer efficiency and coil protection. The wavy fins produce a rippled air flow pattern throughout the coil, creating the air turbulence necessary for efficient heat transfer. The tube size as well as the tube and fin spacing may vary to suit the constraints of design specification.

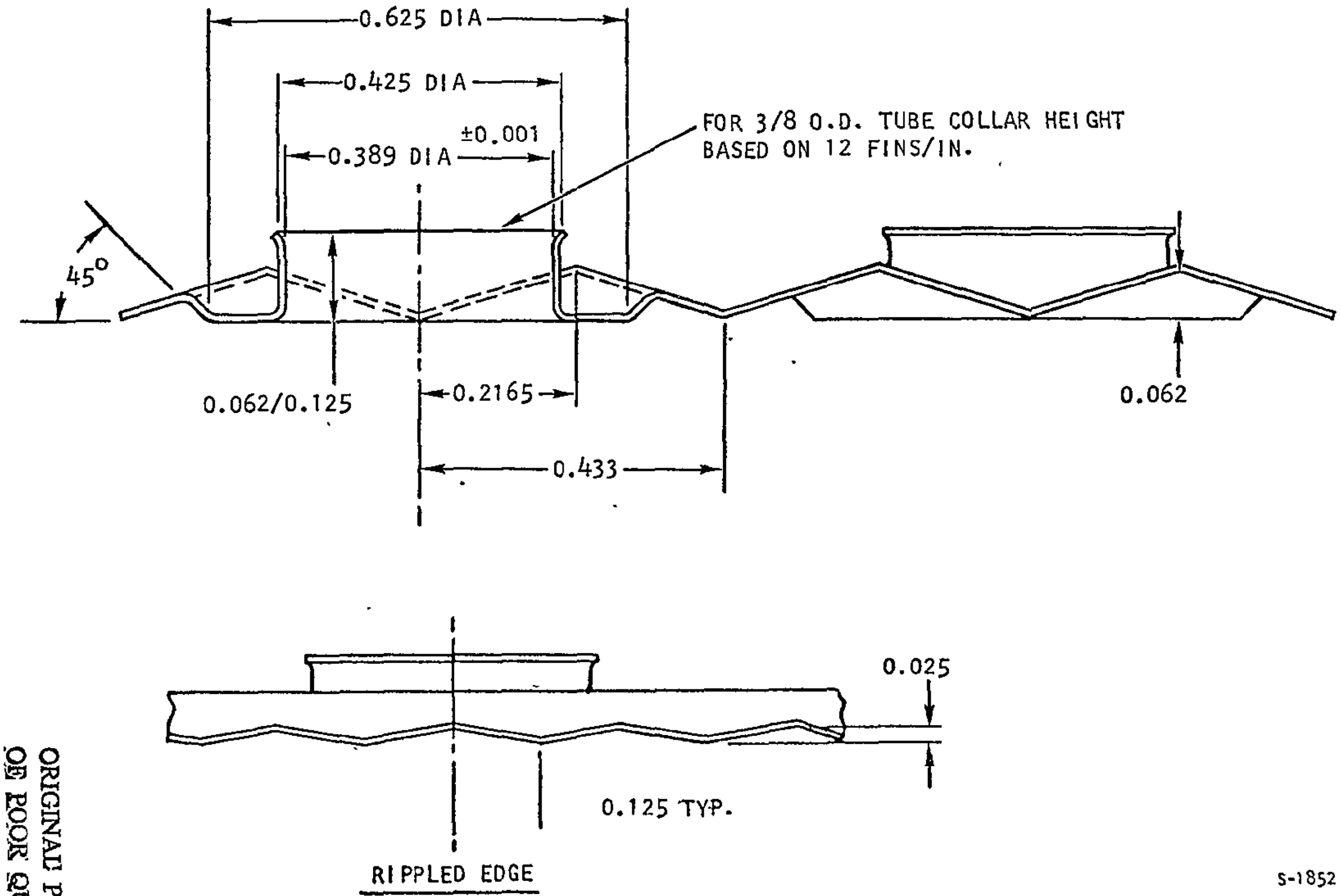

Figure 11-1. Typical Wavv Fin

11.1.2.2 Inner Fin Water Chiller

The inner fin water chiller is a shell-and-tube heat exchanger. The copper tube bundle is contained within a steel shell; water moves over the outside of the tubes and refrigerant moves within. This is a standard heat transfer package that has been marketed by Dunham-Bush for many years.

A sketch of the inner-fin tube arrangement is shown in Figure 11-2. The inner fin is manufactured by inserting a small-diameter copper tube into a copper tube of a larger diameter. A folded spiral fin of copper sheet stock is wedged tightly between the OD of the inner tube and the ID of the outer tube. The folds (points) vary in number around the periphery of the inner diameter; this variation is calculated to produce the most efficient heat transfer.

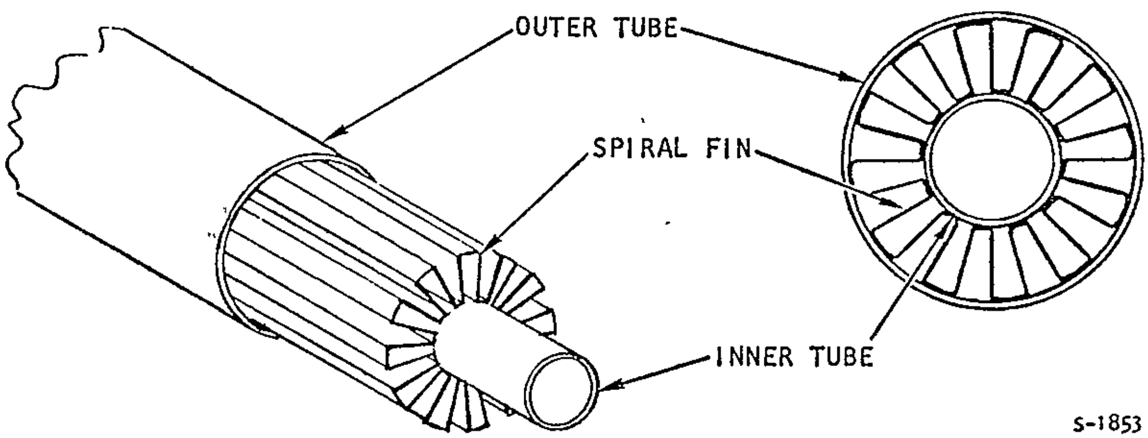

Figure 11-2. Inner-Fin Tube

The inner tube is plugged so that the refrigerant flows only through the finned annulus between the inner and outer tubes while heat is being transferred to or from the water on the outside of the tube.

Water is directed back and forth across the tubes by using baffles. Spacing is calculated to give the most efficient heat transfer on the outside of the tube.

11.1.2.3 Shell-and-Tube Condensers

The third type of heat exchanger is a horizontal low fin tube condenser. In this heat exchanger, water flows through the tubes and R-11 refrigerant condenses on the finned outer surface of the tubes. The condensate drips off of the tubes into the lower portion of the shell, where it can be subcooled by tubes submerged in the R-11 condensate.

Removable water headers are provided for easy cleaning of the inside of the tubes. These exchangers use copper water tubes, steel tube sheets, and cast iron headers constructed and certified to the ASME pressure vessel code. Wolverine type ST low fin tubing with a smooth inner water side surface and a finned outer R-11 condensing surface is used in this design. The fins are necessary to provide efficient condensation of the refrigerant. To provide subcooling, some of the tubes may be immersed in the R-11 condensate.

11.2 HEAT EXCHANGER CHARACTERISTICS

Tables 11-1, 11-2, and 11-3 summarize the sizes of the heating and heating/cooling system heat exchangers of the three types described above. In order to develop a heat pump package common to heating-only and heating/cooling systems, it has been necessary to size the cooling system boiler as well as the other heat exchangers. Further, the use of common heat exchangers for heating and cooling required sizing for both modes of operation.

TABLE 11-1

FINNED-TUBE COIL HEAT EXCHANGERS

Heat Exchanger	No. Required	Tube Dia., in.	Tube Wall Thickness, in.	Finned Tube Length, in.	Fin Width, in.	Rows Deep	Fins Per in.	Fin Thickness, in.
3-ton/60,000 Btu/hr. indoor coil evap/cond R-11 to air	1	0.375	0.016	24	29	8	12	0.006
10-ton/200,000 Btu/hr. indoor coil evap/cond R-11 to air	1	0.375	0.016	48	48	8	12	0.006
25-ton/8×10^5 Btu/hr heater/cooler Water to air	12	0.375	0.016	24	24	8	12	0.006

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TABLE 11-2

INNER-FIN HEAT EXCHANGER CHARACTERISTICS

Heat Exchanger	Shell Dia., in.	Outer Tube		Inner Tube		No. of tubes	Effective tube length, in.	Baffle Spacing in.
		Dia., in.	Thickness, in.	Dia., in.	Thickness in.			
3-ton evaporator/ condenser in water/ R-11 circuit	7.75	0.5	0.028	0.25	0.025	122	32	4
3-ton boiler	3	0.5	0.028	0.25	0.025	19	40	4
10-ton boiler	5.5	0.5	0.028	0.25	0.025	61	40	8
10-ton evaporator	13.5	0.5	0.028	0.25	0.025	381	28	4
25-ton boiler	7.75	0.5	0.028	0.25	0.025	122	45	9
25-ton evaporator	20	0.75	0.032	0.375	0.025	333	49	7

TABLE 11-3

CONDENSER CHARACTERISTICS

Heat Exchanger	Shell dia., in.	Tube dia., in.	Tube Wall Thickness in.	Fins per in.	Fin Thickness, in.	No. of Tubes	Effective Tube Length, in.
25-ton condenser	14	0.75	0.028	19	0.012	96	147

11.3 3-TON/60 KBTUH HEAT PUMP HEAT EXCHANGERS

In this heat pump, two heat exchangers must perform at high effectiveness under two modes of operation: evaporation and condensation (see Figure 5-8). To accommodate both processes in the same heat exchanger requires a compromise solution with careful attention paid to pressure drop in both modes.

11.3.1 Water-to-R-11 Heat Exchanger

For the water-to-R-11 heat exchanger, the Dunham-Bush inner fin tubing provides the means of handling both evaporation and condensation in an efficient manner. Nonstandard water flow routing is required on the shell side, however, to assure subcooling/superheating. This nonstandard routing consists of water entering the shell at the refrigerant discharge and making one pass across the tubes. The water then is piped to the refrigerant inlet end of the shell and flows in a cross-parallel flow arrangement toward the refrigerant discharge end of the shell. This arrangement is depicted below.

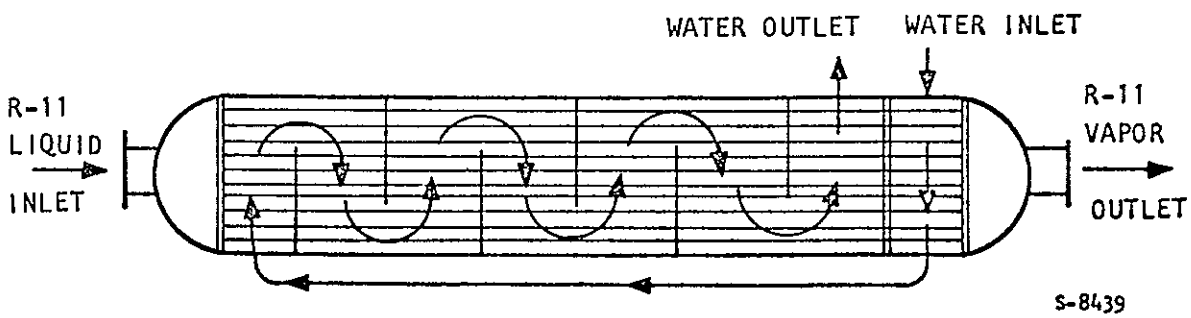

Table 11-4 lists the problem statements (performance requirements) for this unit in the heating and cooling modes of operation. These data were obtained from Figures 5-5 and 5-9. Note that the pressure drop on the R-11 side results in a significant change in evaporating temperature.

The dimensions of this unit are given in Table 11-2.

TABLE 11-4

3-TON/60,000 BTU/HR
WATER TO R-11 HEATING/COOLING UNIT DESIGN CONDITIONS

	Cooling Mode Evaporator	Heating Mode Condenser
R-11 Side		
Q, Btu/hr.	94,900	47,660
Flow, lb/hr	1174	715
Inlet pressure, psia	21.8	9.8
Inlet temperature, °F	117.1	55
Inlet enthalpy, Btu/lb	107.3	32.83
Condensing/evaporating temperature, °F	94.5	55 at inlet; 51 at outlet
Latent heat, Btu/lb	81.8	65.96
Outlet pressure, psia	21.4	9.0
Subcooling/superheat, °F	5	5
Outlet temperature, °F	89.5	56
Outlet enthalpy, Btu/lb	26.5	99.46
Water Side		
Flow, lb/hr	18,808	12,000
Inlet temperature, °F	84.4	60
Outlet temperature, °F	89.5	56°F
Pressure drop, psia	3.5 psia	1.5 psi
Pressure, psia	TBD	TBD

11.3.2 Boiler

The design requirements for the 3-ton heat pump boiler are given in Table 11-5. Boiler characteristics are listed in Table 11-2.

The boiler is a shell and tube unit with water flowing on the shell side directed back and forth across the tubes by using baffles. The baffle spacing is selected to provide the required heat transfer coefficient to the tubes. The R-11 flows through the finned annulus between the inner and outer tubes in a single pass through the inner fin tubes.

11.3.3 Air Coil Heating/Cooling Unit

The finned-tube air heating and cooling coil design conditions for the single-family residence are summarized in Table 11-6. The size of the 3-ton system air coil for the single-family residence is given in Table 11-1. No problems are anticipated in this case to obtain adequate superheating or subcooling.

11.4 10-TON/200-KBTUH HEAT PUMP HEAT EXCHANGER

This system is a variation of the smaller, single-family residence unit. Here again, dual mode of operation is necessary in the heat source/sink heat exchangers.

11.4.1 Water-to-R-11 Heat Exchanger

The water flow configuration in this heat exchanger will be the same as for the smaller unit described above. The problem statements corresponding the heating and cooling modes of operation are listed in Table 11-7. Heat exchanger size and characteristics are listed in Table 11-2.

TABLE 11-5

3-TON HEAT PUMP BOILER REQUIREMENTS

<u>Hot Side</u>	Water
Q: 59,000 Btu/hr	
Flow: 7920 lb/hr	
Inlet temperature: 200°F	
Outlet temperature: 192.5°F	
Inlet pressure: TBD	
Pressure drop: 7.0 psi (max)	
<u>Cold Side</u>	R-11
Flow: 663.6 lb/hr (liquid)	
Inlet density: 90.87 lb/ft^3	
Inlet temperature: 90.9°F	
Inlet pressure: 90 psia	
Inlet enthalpy: 26.8 Btu/lb	
Boiling temperature: 185°F	
Superheat: 5°F	
Outlet pressure: 86.0 psia	
Outlet temperature: 190°F	
Outlet enthalpy: 115.7 Btu/lb	
Latent heat: 66.1 Btu/lb	

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TABLE 11-6

3-TON/60,000 BTU/HR
AIR COIL HEATING/COOLING UNIT DESIGN CONDITIONS

	Cooling Mode Evaporator	Heating Mode Condenser
R-11 Side		
Q, Btu/hr	37,706	59,600
Flow, lb/hr	510.3	715
Inlet temperature, °F	53.0	179.2°F
Inlet pressure, psia	9.39	36.0
Inlet density, lb/ft^3		0.912
Inlet enthalpy, Btu/lb	25.5	116.1
Evaporating/condensing Temperature, °F	53.0 (inlet); 50.46 (outlet)	123.9
Outlet pressure, psia	8.89	35.4
Heat of vaporization, Btu/lb	73.89	82
Outlet enthalpy, Btu/lb	99.39	32.83
Pressure drop, psia	0.50	0.2
Superheat/subcooling, °F	5	5
Air Side		
Flow, lb/hr	5400	5400
Cfm	1200	1200
Inlet db temperature, °F	78	70.6
Inlet wb temperature, °F	67	--
Outlet temperature, °F	57.0 (sat)	116.6
Inlet pressure, psia	14.7	14.7
Pressure drop, in. H_2O	0.36	0.25

TABLE 11-7

10-TON/200,000 BTU/HR
WATER TO R-11 HEATING/COOLING UNIT DESIGN CONDITIONS

	Cooling Mode Condenser	Heating Mode Evaporator
R-11 Side		
Q, Btu/hr	288,351	162,040
Flow, lb/hr	3582	2364.5
Inlet pressure, psia	21.8	9.6
Inlet temperature, °F	111.6	54
Inlet enthalpy, Btu/lb	106.9	30.8
Condensing/evaporating temperature, °F	95	54 at inlet; 50 at outlet
Latent heat, Btu/lb	76.8	67.8
Outlet pressure, psia	21.6	8.8
Subcooling/superheat, °F	5	5
Outlet temperature, °F	89.4	55
Outlet enthalpy, Btu/lb	26.4	99.3
Water Side		
Flow, lb/hr	57,685	36,000
Inlet temperature, °F	84.4	60
Outlet temperature, °F	89.4	55.5
Pressure drop, psia	5.0 psia	2.5 psi
Pressure, psia	TBD	TBD

11.4.2 Boiler

This unit is a shell-and-tube unit using standard inner fin tubes. R-11 is evaporated in the tubes, and water flows through the shell across the tube bundle in a cross-counterflow manner.

The heat exchanger problem statement is in Table 11-8. Heat exchanger characteristics are given in Table 11-2.

11.4.3 Air Coil Heating/Cooling Unit

This coil uses the wavy fin surface geometry on the air side. It is similar to the corresponding unit in the 3-ton 160-KBTUH system but larger in dimension. The design conditions for this heat exchanger in the heating and cooling modes of operation are listed in Table 11-9. Heat exchanger sizes are given in Table 11-1.

11.5 25-TON/600-KBTUH HEAT EXCHANGER

This system differs in that it features a recirculating water loop to carry the heating/cooling effect from the heat pump to the terminal units. Water reversing valves are used throughout so that the heat exchanger processes are not reversed from evaporation to condensation and vice versa.

11.5.1 Condenser

The condenser is a shell and tube unit with condensation occurring in the shell on the surface of low profile finned tubes. Water in the tube serves as the heat sink. Subcooling is achieved by providing additional water-cooled tubes submerged in the liquid refrigerant at the bottom of the shell.

The design conditions for this condenser are listed in Table 11-10. Table 11-3 shows the condenser characteristics.

TABLE 11-8

10 TON/200,000 BTU/HR BOILER

<u>Hot Side</u> Water	
Q: 173,213 Btu/hr	
Flow: 22,500 lb/hr	
Inlet temperature: 200°F	
Outlet temperature: 192.3°F	
Inlet pressure: TBD	
Pressure drop: 7.0 psi (max)	
<u>Cold Side</u> R-11	
Flow: 1955 lb/hr (liquid)	
Inlet density: 90.87 lb/ft^3 (liquid)	
Inlet temperature: 90.7°F	
Inlet pressure: 88.8 psia	
Inlet enthalpy: 26.7 Btu/lb	
Boiling temperature: 184.4°F at outlet	
Superheat: 5°F	
Outlet pressure: 85.3 psia	
Outlet temperature: 189.8°F	
Outlet enthalpy: 115.3 Btu/lb	
Latent heat: 66.1 Btu/lb	

TABLE 11-9

10-TON/200,000 BTU/HR
AIR COIL HEATING/COOLING UNIT DESIGN CONDITIONS

	Cooling Mode Evaporator	Heating Mode Condenser
<u>R-11 Side</u>		
Q, Btu/hr	117,380	200,000
Flow, lb/hr	1626.4	2364.5
Inlet temperature, °F	50.4	170
Inlet pressure, psia	8.84	31.4
Inlet enthalpy, Btu/lb	26.4	114.58
Evaporating/Condensing temperature, °F	50.2 (inlet) 48.1 (outlet)	115.5
Outlet pressure, psia	8.4	30.8
Heat of vaporization, Btu/lb	72.89	74.7
Outlet enthalpy, Btu/lb	98.57	30.8
Pressure drop, psia	0.2	0.2
Superheat/subcooling, °F	2.7 (min)	5
<u>Air Side</u>		
Flow, lb/hr	18,000	22,500
Cfm	4,000	5,000
Inlet db temperature, °F	78	70.6
Inlet wb temperature, °F	67	---
Outlet temperature, °F	57.5 (sat)	107.6
Inlet pressure, psia	14.7	14.7
Pressure drop, in. H_2O	0.34	0.34

TABLE 11-10

25-TON/600,000 BTU/HR
CONDENSER DESIGN CONDITIONS

	Cooling Mode	Heating Mode
R-11 Side		
Q, Btu/hr	690,000	589,635
Flow, lb/hr	8550	6940
Inlet pressure, psia	21.5	35.0
Inlet temperature, °F	111.4	188
Condensing temperature, °F	94.5	123.1
Inlet enthalpy, Btu/lb	107.2	117.1
Outlet pressure, psia	21.4	34.9
Outlet enthalpy, Btu/lb	26.5	32.1
Subcooling, °F	5	5
Water Side		
Flow, lb/hr	135,883	37,500
Inlet temperature, °F	84.4	102.3
Outlet temperature, °F	89.5	117.9
Pressure drop, psia	5.0 (max)	3.0
Pressure, psia	TBD	TBD

11.5.2 Evaporator

The flow configuration through this inner fin water-to-R-11 heat exchanger is similar to that of the smaller units described previously. Heating and cooling mode problem statements are given in Table 11-11. Evaporator characteristics are given in Table 11-2.

11.5.3 Boiler

This boiler is similar to the other units for the smaller systems. Design conditions are listed in Table 11-12, and characteristics are in Table 11-2.

11.5.4 Air-to-Water Terminal Unit Coil

Currently, it is assumed that this system will have 12 terminal units. The design requirements for each unit are listed in Table 11-13. The characteristics of this wavy fin coil are given in Table 11-1.

11.6 SINGLE TUBE HEAT TRANSFER TESTS

Generalized heat transfer and pressure drop data are available in the literature for the design of the various system R-11 phase change heat exchangers (see Appendix B). These data currently are used for this purpose; however, because of the relatively high effectivenesses of the heat exchangers used in the heat pump subsystems, and also because of the sensitivity of the designs, it is planned to conduct single-tube heat transfer and pressure drop tests on inner fin tubes of the sizes used in the design of the heat pump heat exchangers.

The purpose of these tests is to verify performance prediction and to ascertain the designs prior to fabrication. This approach is common practice in the design of high-efficiency heat exchangers and obviates costly iterations later in the program, as well as schedule slips.

A schematic of the test rig is shown in Figure 11-3. The scope of the test program is delineated in Tables 11-14 and 11-15.

TABLE 11-11

25-TON/600,000 BTU/HR
EVAPORATOR* DESIGN CONDITIONS

	Cooling Mode	Heating Mode
R-11 Side		
Q, Btu/hr	295,300	462,551
Flow, lb/hr	4095	6940
Inlet temperature, °F	8.6	53.4
Inlet pressure, psia	8.5	9.37
Inlet enthalpy, Btu/lb	26.5	32.07
Saturation temperature, °F	48.8 (inlet); 45.7 (outlet)	53.4 (inlet); 47.2 (outlet)
Outlet enthalpy, Btu/lb	98.56	98.72
Heat of vaporization, Btu/lb	71.47	66.21
Outlet pressure, psia	8.1	8.37
Pressure drop, psi	0.5	1.0
Superheat, °F	4.3	4
Water Side		
Flow Rate, lb/hr	37,500	80,000
Inlet temperature, °F	58	60
Outlet temperature, °F	50	54.2
Inlet pressure, psia	TBD	TBD
Pressure drop, psi	1.5	4.3

* Dual mode in Nashville only

TABLE 11-12

25-TON BOILER

<u>Hot side</u> Water	
Q: 394,700 Btu/hr	
Flow: 50,700 lb/hr	
Inlet temperature: 200°F	
Outlet temperature: 192.2°F	
Inlet pressure: TBD	
Pressure drop: 6.0 psi (max)	
<u>Cold Side</u> R-11	
Flow: 4455 lb/hr	
Inlet density: 90.87 lb/ft^3 (liquid)	
Inlet temperature: 90.8°F	
Inlet pressure: 89.6 psia	
Inlet enthalpy: 26.7 Btu/lb	
Boiling temperature: 188.5°F (inlet); 189.6°F (outlet)	
Heat of vaporization: 66.18 Btu/lb	
Superheat: 5°F	
Outlet enthalpy: 115.3 Btu/lb	
Outlet pressure: 85.6 psia	
Outlet temperature: 189.6°F	

TABLE 11-13

25-TON/600,000 BTU/HR
AIR HEATER/COOLER DESIGN CONDITIONS
(12 Required)

Water Side	Heating Mode	Cooling Mode
Q, Btu/hr		
Flow Rate, lb/hr	48,500	24,600
Inlet temperature, °F	3,125	3,125
Outlet temperature, °F	117.9	50
Pressure, psia	102.3	58
Pressure drop, psi	TBD	TBD
Air Side	1.5	1.5
Flow rate, lb/hr		
Flow rate, cfm	5,420	3,750
Inlet db temperature, °F	1,200	830
Inlet wb temperature, °F	70.5	78
Outlet temperature, °F	N/A	67
Pressure, psia	107.7	57.5 (sat)
Pressure drop, in. H_2O	14.7	14.7
	0.29	0.28

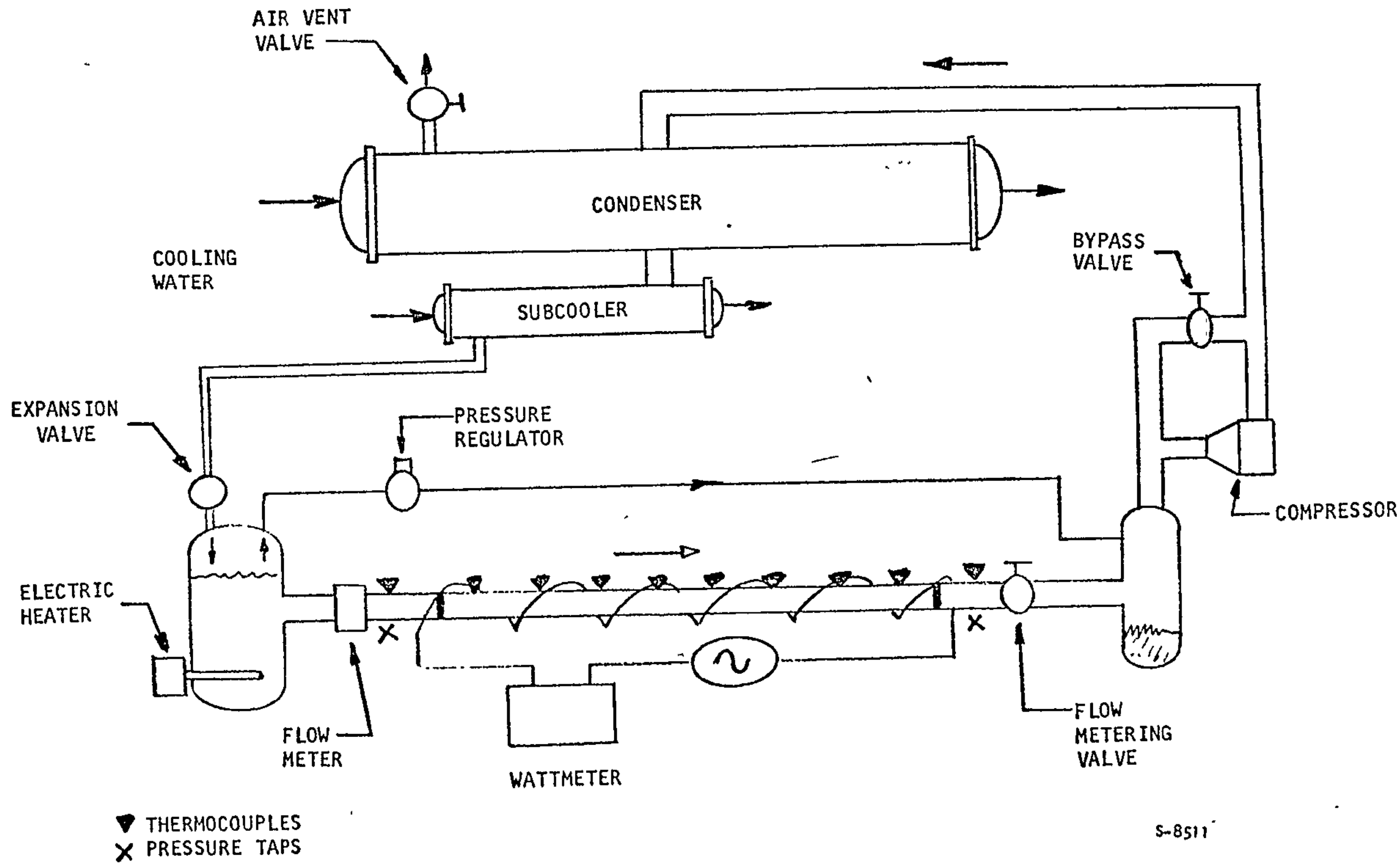

Figure 11-3. Single Tube Boiling Test Setup for Freon R-11 in a Dunham-Bush Inner Fin Tube

TABLE 11-14

SINGLE TUBE R-11 BOILING TEST CONDITIONS
DUNHAM-BUSH INNER FIN TUBE MODEL 34-38 (CIC)
60 INCHES LONG

Inlet R-11		R-11 Flow Rate, lb/hr	Heat Flow, watts	Estimated ΔP, psi	Estimated ΔT Wall to Fluid, °F
Press, psia	Temp, °F				
8.6	48.9	12.3	291.0	0.5	3.8
9.47	53.4	20.84	490.7	0.9	4.4
9.7	54.5	30.0	706.4	1.8	5.0

TABLE 11-15

SINGLE TUBE R-11 BOILING TEST CONDITIONS
DUNHAM-BUSH INNER FIN TUBE MODEL 12-14 (FT-45A)
40 INCHES LONG

Inlet R-11		R-11 Flow Rate, lb/hr	Heat Flow, watts	Estimated ΔP, psi	Estimated ΔT Wall to Fluid, °F
Press, psia	Temp, °F				
88.6	90.8	23.0	596.4	2.0	2.4
89.6	90.8	35.0	907.6	4.0	3.7
91.6	90.8	52.5	1361.34	8.0	4.8
9.8	55.0	5.86	137.7	0.8	2.7
9.6	54.0	6.206	146.0	0.9	3.0
9.4	53.0	4.0	94.1	0.45	2.2
10.25	57.0	8.0	188.2	1.4	3.8

The condenser in the test setup should be capable of rejecting 7500 Btu/hr of heat to 75°F entering water while condensing 90 lb/hr of Freon R-11 at 146°F and 50 psia. The subcooler should be capable of rejecting 1290 Btu/hr of heat to 75°F water while cooling 90 lb/hr of Freon R-11 from 146°F to 80°F. The pressure regulator valve should be capable of a Freon R-11 vapor flow of 40 lb/hr at an inlet pressure of 13.0 psia and inlet temperature of 69°F. The expansion valve should be capable of a Freon R-11 liquid flow ranging from 9 lb/hr to 120 lb/hr with a pressure drop of 20 psi to 80 psi. The flow meter and flow metering valve should be selected to operate within the range shown in Tables 11-14 and 11-15. The compressor bypass valve should be capable of the full compressor flow rate.

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12. SYSTEM/SUBSYSTEM TRADE STUDIES

12.1 GENERAL

System-level investigations were conducted to optimize the configuration of the systems in the heating and cooling modes of operation and to identify alternate approaches that could result in improved performance and/or reduced cost. These analyses were extended to cover heat pump design parameter optimization and fluid selection studies to verify the selections made prior to contract award.

The AiResearch system performance prediction computer program was used in the evaluation of the various approaches. Performance was determined over a complete year of operation using as a basis the building models supplied by NASA in RFP AP32-75-404 and the NASA weather tapes for Nashville and Madison. Performance and cost data obtained for the alternate configurations considered were compared to the baseline data furnished in the AiResearch proposal for the three sizes of heating and heating/cooling systems. These baseline data are presented in Tables 12-1 and 12-2 for ease of reference.

The sensitivity analyses covered the following topics:

(a) Evaluation of performance/cost gains afforded by the preheater in the heating and heating/cooling subsystems

(b) Determination of the penalties inherent with the use of the interchanger in the solar collector loop

(c) System sensitivity to heat losses

(d) Optimum water tank temperature control set point in the heating mode of operation

TABLE 12-1

PROPOSED HEATING SYSTEMS PERFORMANCE SUMMARY

	Single-Family Residence	Multifamily Residence	Commercial Application
<u>System Features</u>			
Collector area, sq ft	1000	10,000	22,500
Thermal energy storage tank, gal	1200	18,000	36,000
Heat pump capacity, Btu/hr	80,000	800,000	2,000,000
Auxiliary heater capacity, Btu/hr	80,000	800,000	2,000,000
Domestic hot water			
Storage tank, gal	50	650	100
Recovery rate, Btu/hr	60,000	750,000	100,000
<u>Yearly Performance</u>			
Residence load, 10^6 Btu	213	2130	4650
Hot water load, 10^6 Btu	30.7	384	384
Solar contributions, %	61.2	63.4	59.3
Auxiliary thermal energy contribution, %	33.7	30.6	32.9
Auxiliary electrical energy contribution, %	5.1	6.0	7.8
Total energy expenditure			
Thermal (fuel oil or gas), 10^6 Btu	127	1100	2362
Electrical, kw-hr	7670	87,900	217,000
Ultimate energy saving, 10^6 Btu/year	162	1780	2900
Present value benefit, $	3600	59,000	154,000
(20-year, residential; 25-year, commercial)			

TABLE 12-2

PROPOSED HEATING/COOLING SYSTEMS PERFORMANCE SUMMARY

	Single-Family Residence	Multi-family Residence	Commercial Application
System Features			
Collector area, sq ft	1000	10,000	20,000
Energy storage tank, gal	1800	18,000	48,000
Heat pump capacity			
Heating, Btu/hr	80,000	800,000	2×10^6
Cooling, tons	3	25	75
Auxiliary heater capacity, Btu/hr	80,000	800,000	2×10^6
Domestic hot water			
Storage, gal	50	650	100
Recovery rate, Btu/hr	60,000	750,000	100,000
Yearly Performance			
Residence loads			
Heating, 10^6 Btu	184	1840	4120
Cooling, 10^6 Btu	34.7	294	1090
Hot water load, 10^6 Btu	30.7	384	384
Solar contribution			
Heating, %	67	67	62.6
Cooling, %	75	65	58
Auxiliary thermal energy contribution			
Heating, %	25.1	24.3	27.3
Auxiliary electrical power contribution			
Heating, %	7.9	8.7	10.1
Cooling, %	25	35	42
Total energy expenditure			
Thermal (fuel oil or gas), 10^6 Btu	83.2	772	1753
Electrical, kw-hr	8670	94,185	247,610
Ultimate energy saving, 10^6 Btu	208	2080	3600
Present value benefit, $ (20-year, residential; 25-year, commercial)	3000	71,000	200,000

(e) Optimization of condensing temperature in the cooling mode of operation

(f) Working fluid selection studies

12.2 EVALUATION OF PREHEATER BENEFITS

12.2.1 System Implication

The preheater was incorporated into the proposed design to reduce the load on the heat pump whenever the storage water tank temperature exceeds the temperature of the residence. In this manner, solar thermal energy is used directly without processing through the heat pump. This presents an advantage particularly in the fall and spring seasons when the residence loads are low and the water tank temperature is relatively high.

On the other hand, elimination of the preheater results in the following significant advantages:

(a) Lower initial system cost due to elimination of the preheater itself, two flow control valves, and piping associated with preheater operation

(b) Simplification of system control logic

(c) Lower fan and pump power

12.2.2 System Analysis Data

The performance of the three sizes of heating systems was determined for the entire year, and the present value benefit was determined in comparison to the baseline system data of Table 12-1. Similar data were obtained for the single-family residence. These data are summarized in Table 12-3.

TABLE 12-3

ELIMINATION OF PREHEATER

PERFORMANCE AND COST IMPACT

	Heating Systems			Heating/Cooling System
	Single-Family	Multifamily	Commercial	Residential
$Q_{Solar\ total}$	-0.7%	-1.2%	-1.0%	-1.3%
$Q_{Fuel\ oil}$	--	+1%	+0.9%	--
Electrical energy	+2.0%	-1.2%	-2.5%	+5%
Cost Δ, $				
Fixed cost	-500	-5200	-11,300	-500
Present value benefit*	+390	+5060	+12,100	+340

*Positive sign indicates lower life-cycle cost

As expected, elimination of the preheater results in lower solar energy utilization and higher auxiliary energy usage; however, the overall effect in terms of energy is relatively small. This is due to the very high performance of the heat pump under conditions prevailing when the preheater is thermally useful. For example, with a storage water tank temperature of 80°F and a heat pump operating at 60 percent of rated capacity, the heat pump thermal COP is approximately 9. It follows that in the off-season only a little electrical power is necessary for heat pump operation. This power expenditure is partially offset by the savings in pumping power necessary to (1) overcome the air pressure drop through the preheater, and (2) assure water flow through the preheater, lines, and valves.

In all cases, a cost benefit is realized through elimination of the preheater due primarily to a significant reduction in the initial and installation cost of the heat pump (about 13 percent for the heating systems).

12.2.3 Recommendations

The preheater and associated hardware are eliminated from the heat pump for all system sizes.

12.3 INTERCHANGER PENALTIES

12.3.1 System Implications

The arrangement of the proposed collector loop is shown in Figure 12-1. The collector loop is closed and interfaces thermally with the remainder of the system through an interchanger. The interchanger is a steel unit and is incorporated into the design to permit the use of low-cost aluminum collector plates while maintaining noncorrosive conditions in the relatively small volume of collector fluid in that loop.

The major disadvantages of the interchanger are:

(a) Thermal energy degradation resulting in larger collector areas or conversely in lower tank temperatures

(b) Higher electrical power usage to drive the interchanger pump

(c) Additional hardware costs for the interchanger itself and the interchanger pump

12.3.2 System Analysis Data

Table 12-4 summarizes the effect of removing the interchanger from the system by comparison to the baseline system proposed. The use of the interchanger results in disadvantages with respect to all system evaluation criteria. In particular, the interchanger contributes a significant portion

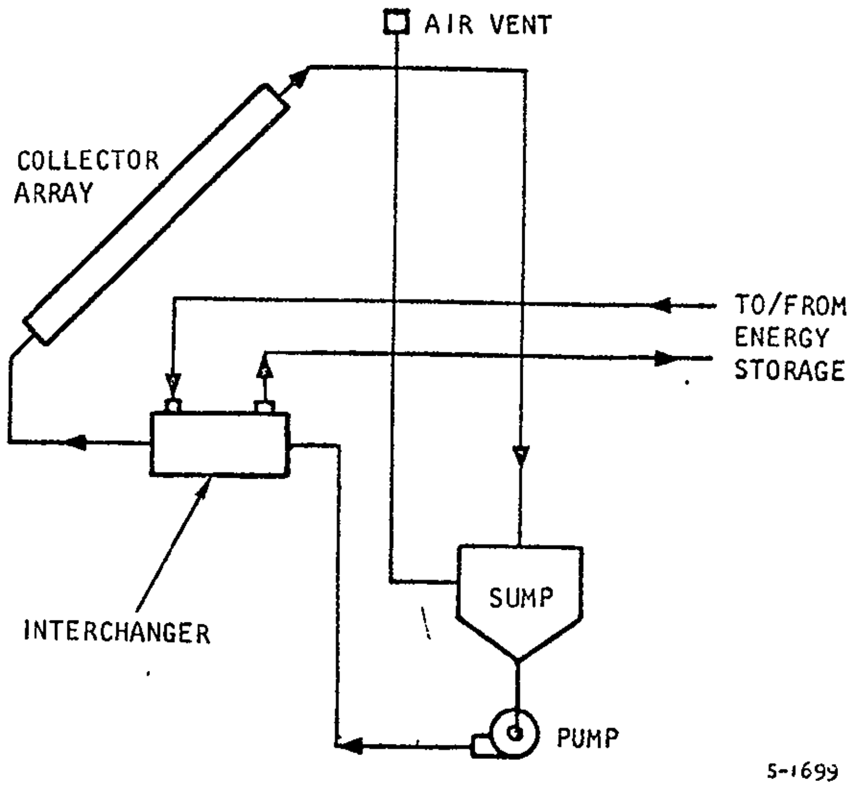

Figure 12-1. Collector Loop Arrangement

of the total electrical energy used by the system and also represents an initial system cost increase of 4 to 6 percent for the heating system and 2 to 3 percent for the heating/cooling system.

12.3.3 Recommendations

Sizeable savings can be effected by eliminating of the interchanger. This aspect of system design will be considered in evaluating candidate collectors. If a particular collector needs an interchanger for corrosion protection, then the values shown in Table 12-4 will be used as a penalty for the particular unit.

TABLE 12-4

ELIMINATION OF INTERCHANGER

PERFORMANCE AND COST PENALTY

	Heating Systems			Heating/Cooling Systems		
	Single-Family Residence	Multifamily Residence	Commercial	Single-Family Residence	Multifamily Residence	Commercial
$Q_{Solar\ total}$	+0.2%	+1.5%	+1.4%	+1.1%	+0.8%	+0.9%
$Q_{Fuel\ oil}$	-1.5%	-2%	-1.8%	--	-7.0%	-1.8%
Electrical energy	-6.9%	-5.9%	-8.9%	-4.8%	-6.8%	-7.9%
Cost Δ, $						
Fixed cost	-400	-2400	-3000	-800	-2500	-4200
Present value benefit*	+900	+6800	+16,400	+900	+8600	+12,200

*Positive sign indicates lower life-cycle cost.

12.4 SENSITIVITY TO HEAT LOSSES

12.4.1 System Implications

In the development of the proposed system design, subsystem cost trade studies were conducted to optimize insulation thicknesses on the system lines and water storage tank. Figures 12-2 and 12-3 present the results of these studies. Estimates were made of all system line sizes, and tank geometry models were used to permit mechanization of the system heat losses for any tank size and operating temperature levels. Using these correlations, the system heat losses were calculated to be about 3.3 percent of the heat collected.

Insulation optimization at the subsystem level does not account for all parameters contributing to the overall performance of the system. System-level investigations were conducted to determine performance sensitivity in terms of heat losses.

12.4.2 System Analysis Data

The analysis involved modifying the baseline heat loss model so that the heat loss characteristics of the lines and tanks would be doubled. The multi-family residence heating/cooling system was used as a basis for the analysis. The computations were run for the entire year using available weather data.

A comparison of the baseline (or proposed) system performance with that of a system where heat losses are doubled is presented in Table 12-5.

As anticipated, slightly more solar energy is collected when the heat losses are higher. However, this increase is not sufficient to make up for the additional losses; and more auxiliary energy is used particularly during the cooling mode of operation. This is due to the higher temperature difference between the water storage tank and outside ambient during the summer months.

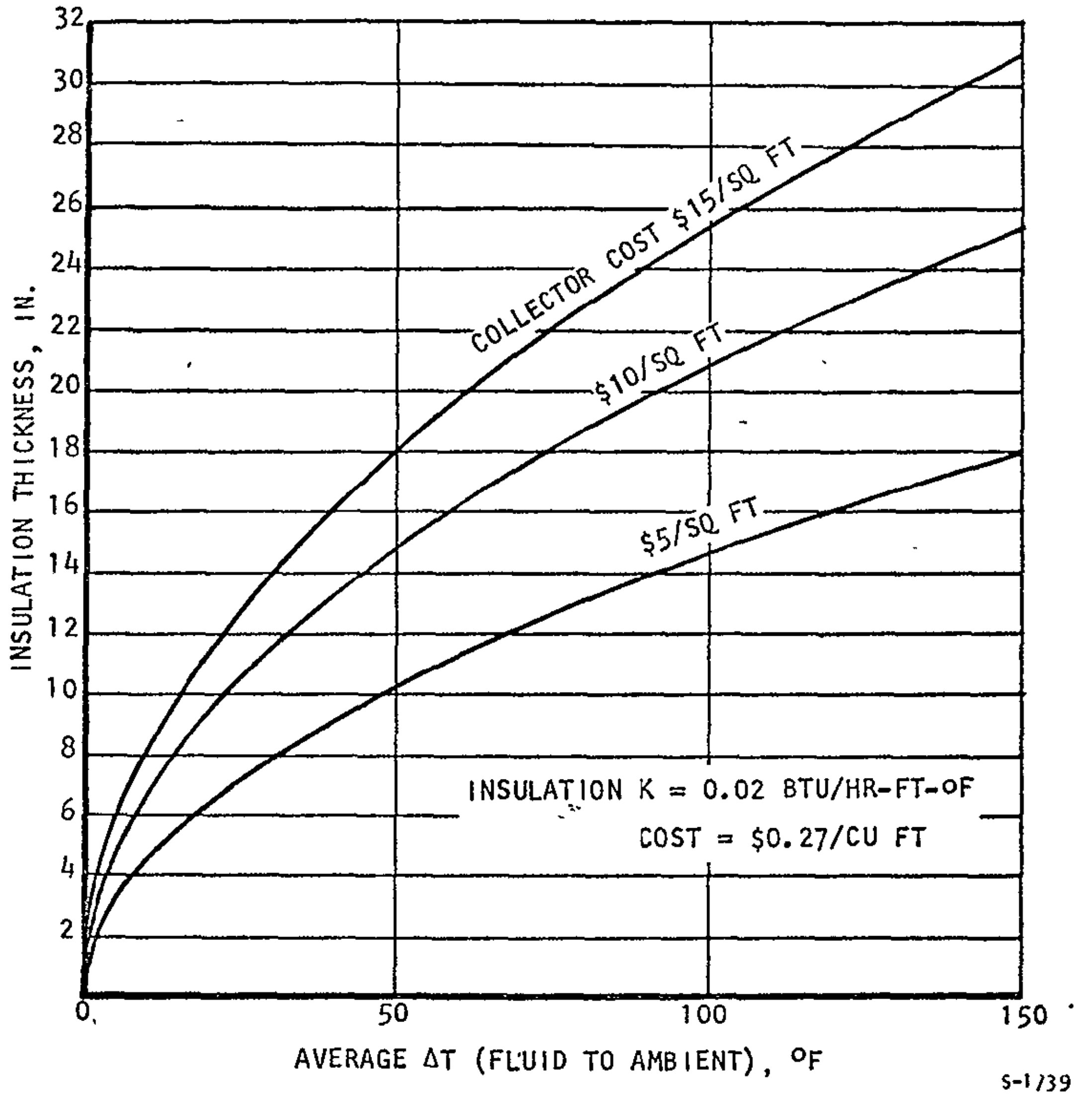

Figure 12-2. Optimum Insulation Thickness for Storage Tank

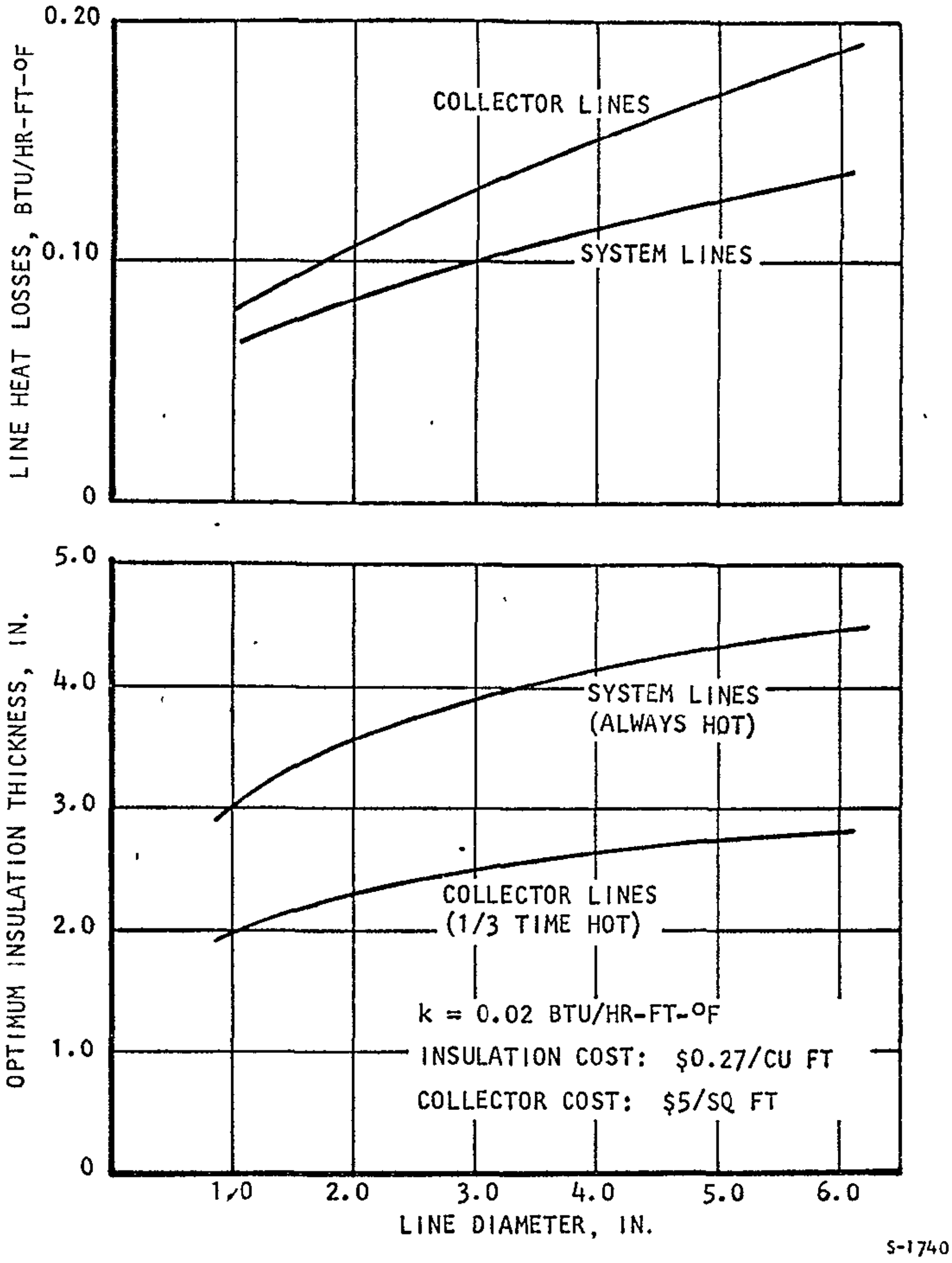

Figure 12-3. Parametric Line Data

TABLE 12-5

SYSTEM SENSITIVITY TO HEAT LOSSES

Multifamily residence heating/cooling system
Capacity: 25-ton cooling; 800 KBtu/hr heating
No preheater
Collector area: 10,000 sq ft
Storage tank capacity: 18,000 gal

YEARLY PERFORMANCE

	Baseline Heat Losses	Twice Baseline Heat Losses
$Q_{collected}$, 10^6 Btu/yr	1820	1850 (+1.6%)
Q_{cost}, 10^6 Btu/yr	47.9	93.8 (+96%)
Heating load, 10^6 Btu/yr		
Residence	1840	
DHW	383	
Cooling load, 10^6 Btu/yr	294	
Q_{solar}, 10^6 Btu/yr		
Residence heating	1210	1210
DHW	261	259
Residence cooling	294	294
Auxiliary heating, 10^6 Btu/yr		
Gas-oil consumed	543	548.3 (+1%)
Auxiliary, electrical energy	209	205.7 (-1.6%)
Auxiliary cooling, kw-hr/yr	1590	1750 (+10%)
Total electrical energy, kw-hr/yr		
Heating	80,500	80,600
Cooling	15,790	16,050
Total	96,290	96,650 (+0.4%)

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12.4.3 Recommendations

Insulation optimization will be performed using system-level data as a basis for cost trade studies.

12.5 WATER STORAGE TANK TEMPERATURE CONTROL

12.5.1 System Implications

The proposed heating systems feature a control to limit heat pump operation to stored water temperature above 40°F. This will obviate freezing problems in the evaporative heat exchangers of the heat pumps.

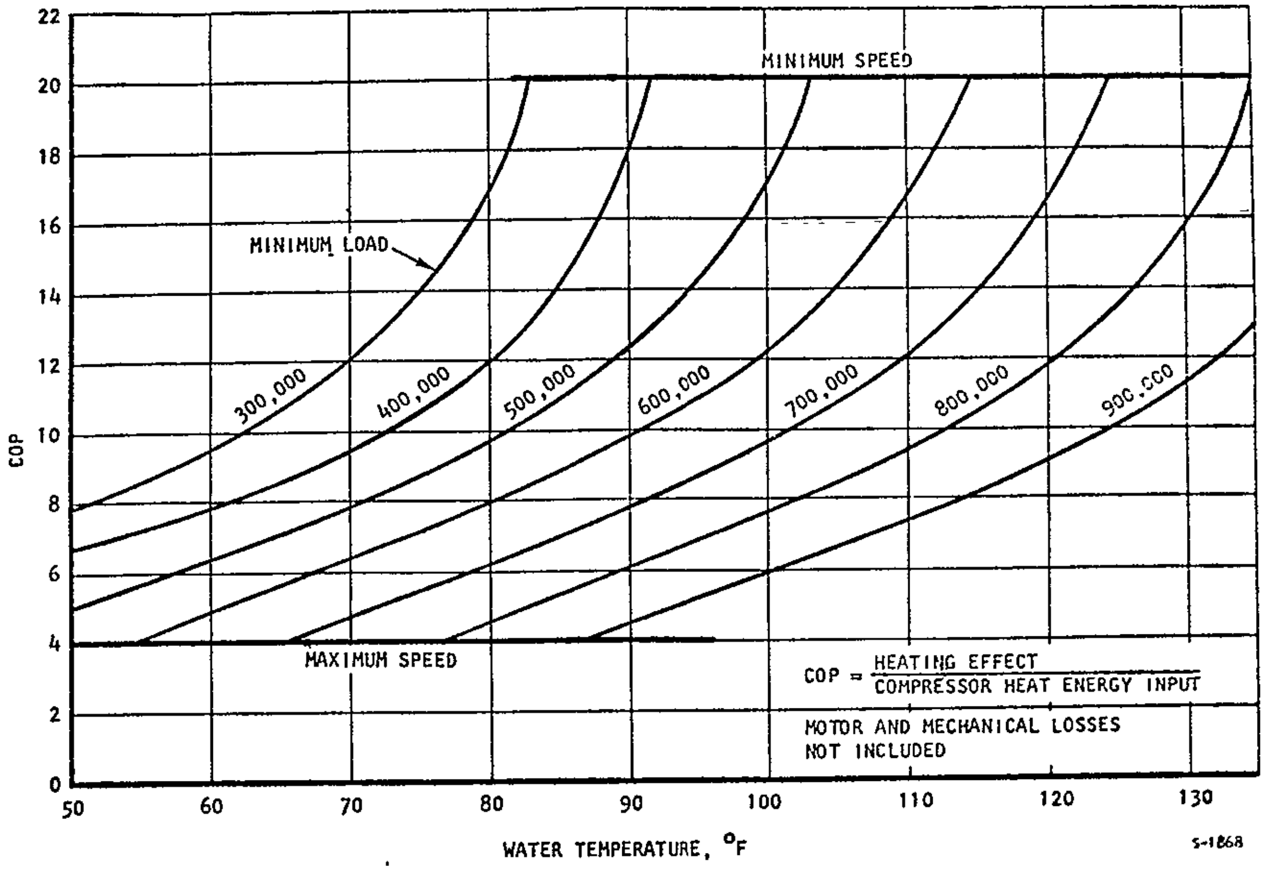
Should a higher minimum water temperature set point be selected, the heat pump may operate at higher capacity and also at higher COP. Operation in this manner would result in higher collector temperatures and reduced efficiencies, but also in potential reduction in electrical energy usage.

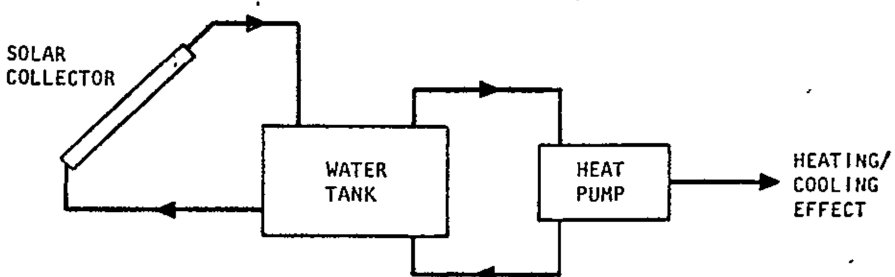
12.5.2 System Analysis Data

The performance of the multifamily heating system was calculated for minimum tank temperature set points from 40° (baseline) to 70°F. The Madison weather tape was used for this purpose together with the residence model specified in the NASA RFP.

The results of this investigation are summarized in Figure 12-4. As shown, the performance of the solar collector deteriorates rapidly as the tank temperature increases. Considerably less heat is processed through the heat pump and less electrical energy is expended. On the other hand, the thermal auxiliary energy used increases rapidly as the tank is controlled at higher temperature.

The overall result is a significant increase in operating costs with a reduced economic benefit.





MADISON
800,000 BTU/HR SYSTEM

MINIMUM WATER TEMPERATURE	40°F	50°F	60°F	70°F
Q SOLAR, 10^6 BTU/YEAR	1474	1396 (-5%)	1298 (-12%)	1212 (-18%)
Q FUEL OIL, 10^6 BTU/YEAR	794	918 (+15%)	1044 (+31%)	1161 (+46%)
ELECTRICAL ENERGY, KW-HR/YEAR	82,300	69,800 (-15%)	57,400 (-29%)	48,400 (-41%)
ULTIMATE ENERGY SAVINGS	BASELINE	+0.35%	+1.1%	+2.8%
PRESENT VALUE BENEFIT, Δ$*	BASELINE	-4500	-9570	-15,300

S-8457

*NEGATIVE VALUE INDICATES HIGHER LIFE-CYCLE COST.

Figure 12-4. Effect of Water Tank Temperature Control

12.5.3 Recommendation

Water storage tank temperature should be controlled at a minimum level determined by water freezing considerations.

12.6 OPTIMIZATION OF CONDENSING TEMPERATURE IN COOLING MODE

12.6.1 System Implications

In the cooling mode of operation, condensing temperature is the single most important system parameter, since it affects both the Rankine power loop efficiency and the refrigeration loop COP.

In designing the proposed system, subsystem-level trades were conducted to optimize condensing temperature using design-point operating parameters. A temperature of 92°F was selected on that basis. This low condensing temperature imposes severe constraints on the design of both the condenser and the cooling tower.

Examination of system performance data for the entire year indicates that the system seldom operates at rated conditions; thus, the subsystem trades conducted using the design point parameters with respect to water source temperature (200°F) and ambient conditions (95°F dry bulb, 78°F wet bulb) may not yield the optimum heat pump configuration. Instead, system-level performance evaluation should be conducted for optimization of condensing temperature.

12.6.2 System Analysis Data

The single-family heating/cooling system was used to determine the effect of condensing temperature on overall system performance. The NASA-supplied residence model and the Nashville weather tapes also were used. The off-design performance of the heat pump designed with a condensing temperature of 94°F was characterized over the entire range of ambient dry- and wet-bulb temperature conditions and water-source temperature level.

The results of the analyses are shown in Table 12-6 for a complete year of operation. The data are presented for the same solar collector and storage tank capacity. The data show a small overall advantage in increasing the condensing temperature. Although more auxiliary electrical energy is used to drive the turbocompressor motor, this is more than offset by the power savings realized by selection of the cooling tower performance requirements.

Figure 12-5 illustrates the cooling circuit temperatures and flow rates at design point for the two condensing temperatures investigated. As shown, the approach temperature at the cooling tower is increased from 5° to 6°F corresponding to a significant reduction in the size of the cooling tower. Further, the water inlet temperature at cooling tower inlet is increased by 2°F.

In terms of condenser design, the UA requirements in both cases are about the same.

12.6.3 Recommendations

The design point condensing temperature is increased from 92° to 94°F to relax the cooling tower performance requirements. Although system COP at design point drops from 0.62 to 0.59, the overall system performance for the entire cooling season is not affected significantly.

12.7 WORKING FLUID SELECTION

12.7.1 System Implications

R-11 is a relatively low-pressure refrigerant with desirable thermodynamic properties. However, it is not commonly used in the refrigeration industry primarily because of the large heat exchanger sizes resulting from its low density. In an effort to reduce heat exchanger sizes and heat pump cost, detailed investigations were conducted to identify alternate higher pressure

TABLE 12-6

EFFECT OF CONDENSING TEMPERATURE

Single-family residence heating/cooling system
Collector area: 100 ft^2
Storage tank capacity: 1800 gal

Condensing temperature	92°F	94°F
Loads, 10^6 Btu/yr		
Heating load		
Residence		184
DHW		30.7
Cooling load		34.7
Q_{Solar}, 10^6 Btu/yr		
Heating		
Residence	121	121
DHW	20.9	20.9
Cooling	37.6	37.6
Auxiliary heating, 10^6 Btu/yr		
Gas/oil	51.5	51.5
Electrical energy	21.3	21.3
Auxiliary cooling, kw-hr	293	316
Total electrical energy, kw-hr		
Heating	7290	7230
Cooling	1853	1756
Total	9143	8986 (-1.7%)

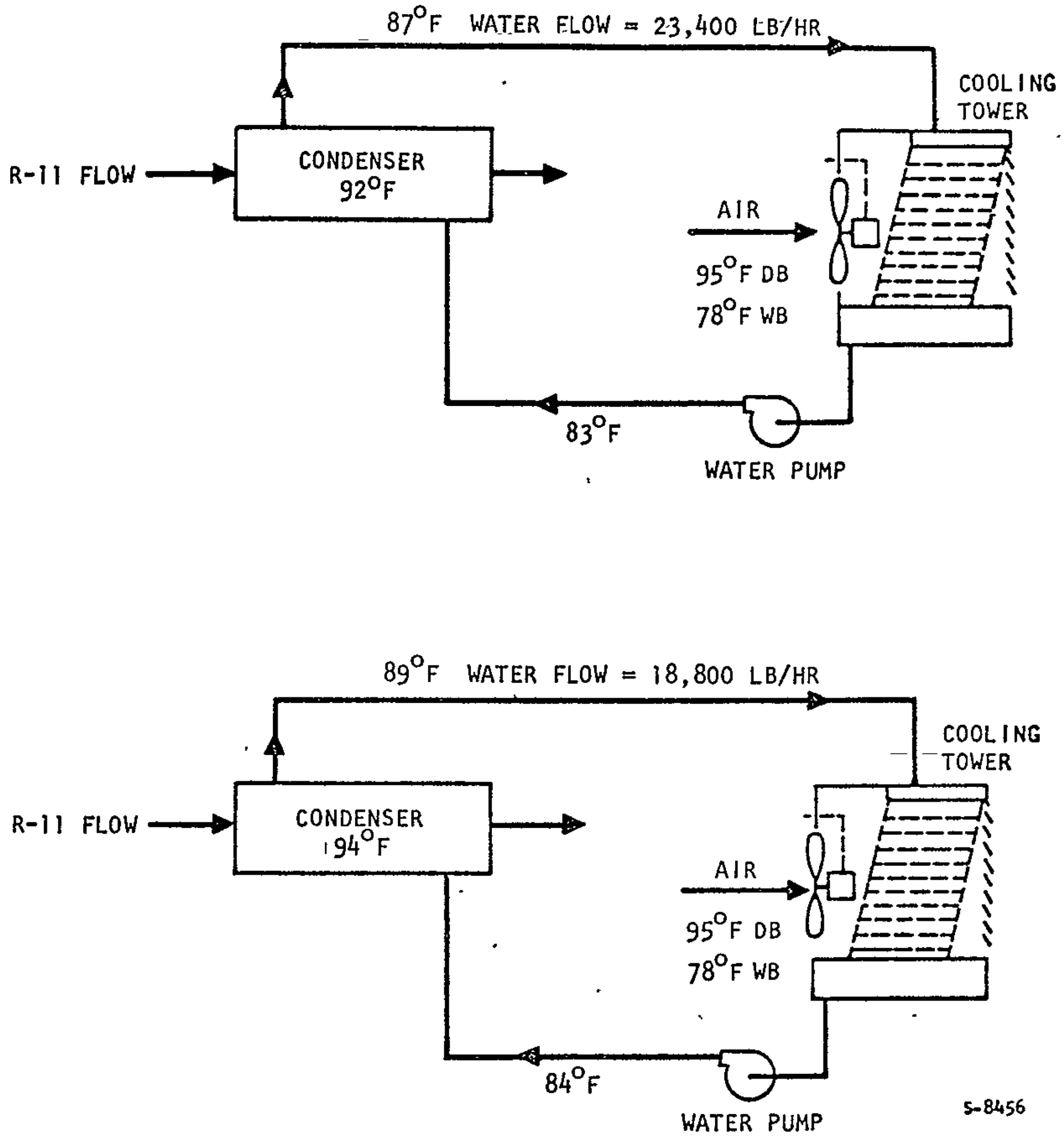


Figure 12-5. Comparison of Condenser Cooling Circuits

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working fluids. The two fluids considered were R-12 and R-114. Pertinent thermodynamic properties of R-11, R-12, and R-114 are listed in Table 12-7. Both the heating and cooling modes of operation were investigated. The results of these investigations are summarized below.

12.7.2 Analysis Data

12.7.2.1 R-12 Working Fluid

In the heating mode of operation, R-12 could be used to advantage for the large multifamily and commercial systems. For the small 60 KBtu/hr heat pump, the low volumetric flow results in very small centrifugal compressor passages and thus prohibitively low compressor efficiencies. The lower system size limit to achieve compressor efficiencies on the order of 70 percent is about 150 to 200 KBtu/hr.

In the cooling mode of operation, the low latent heat of R-12 at temperature levels around 200°F yields large working fluid flow. In addition, the characteristic vapor pressure of this refrigerant near the critical temperature is such that about 30°F of superheat would be required at boiler outlet. It follows that with a heat source temperature of about 180°F, the boiling temperature has to be reduced to about 150°F and the turbine pressure ratio drops to about 2:1. In terms of the Rankine power loop, the high flows combined with the low turbine pressure ratios will result in unacceptable power loop performance and overall system COP.

For these reasons, R-12 was rejected as a cooling system working fluid. Commonality between the heating and heating/cooling system hardware eliminates R-12 from further consideration.

TABLE 12-7

PROPERTIES OF CANDIDATE REFRIGERANTS

Refrigerant	R-11	R-12	R-14
Chemical formula	CCl_3F	CCl_2F_2	$C_2Cl_2F_4$
Molecular weight	137.38	120.93	170.94
Boiling temperature at atmospheric pressure, °F	74.9	-21.6	38.8
Critical temperature, °F	388.4	233.6	294.3
Density saturated vapor at 45°F, lb/cu ft	0.2046	1.406	0.5525
Latent heat at 45°F, Btu/lb	80.1	63.6	58.0
Latent heat at 200°F, Btu/lb	64.8	32.1	40.2
Saturated pressure at 200°F, psia	105.5	430.1	178.4
Saturated pressure at 70°F, psia	13.35	84.9	27.26

12.7.2.2 <u>R-114 Working Fluid</u>

R-114 offers the advantages of higher pressure and higher density by comparison with R-11. On the other hand, the latent heat of R-114 is considerably lower than that of R-11 (58 Btu/lb vs 80 Btu/lb at 45°F). Thermodynamically, R-114 is a much inferior working fluid. Overall system COP at design point is 0.43. This compares to a COP of 0.62 using R-11. The net results in terms of heat pump design are:

(a) Much increased flow rates through the Rankine power loop and the refrigeration loop. This results in heat exchanger pressure drop characteristics as measured by ($\dot{w}2/\rho$) about the same as obtained with R-11.

(b) Increased heat exchanger (boiler and condenser) loads due to the significantly lower COP obtained with R-114. The overall effect is larger heat exchanger sizes.

(c) Increased collector and cooling tower sizes due to the lower COP's. This in itself is sufficient to eliminate R-114 from further considerations.

12.7.2.3 Recommendations

The selection of R-11 as the optimum working fluid for the solar heating and cooling systems has been confirmed. Heat pump subsystem and equipment preliminary design will proceed on this basis.

C-3

APPENDIX A

RATIONALE FOR SELECTION OF 3-, 10-, AND 25-TON SIZES FOR HEATING/COOLING SUBSYSTEM DEVELOPMENT

APPENDIX A

RATIONALE FOR SELECTION OF 3-, 10-, AND 25-TON SIZES FOR HEATING/COOLING SUBSYSTEM DEVELOPMENT

INTRODUCTION

Per NASA request, AiResearch and Dunham-Bush conducted a market survey and analysis to determine the most marketable heating/cooling subsystem sizes. Figures A-1 through A-13 (presented at the end of this appendix) show the projections for total industry air conditioning sales by size through 1986. The year 1983 was selected as a representative year in which solar systems might be readily available for installation. The projected unit sales for 1983 for each capacity size were multiplied by a typical unit price to arrive at a total dollar market volume for each size. This information is presented in Table A-1.

SINGLE-FAMILY UNITS

The data of Table A-1 indicate that a 3-ton unit has the largest potential for single-family application. Projected annual sales for the year 1983 show a total of 235,000 units for a total dollar value of $164,500,000. In comparison, total sales for 5-ton units are expected to reach 103,000 for $113,300,000 annually.

The 3-ton size was therefore selected for demonstration purposes as being the most representative of this type of equipment. Also, there should be the greatest number of prospective sites for installing 3-ton systems. Data obtained on a 3-ton system will be representative of those for other sizes of single-family units.

A single-package integral system presents the best arrangement for proving the concept. This type system can be manufactured entirely in the contractor's plant. It would incorporate as standby heating equipment the gas-fired furnace

TABLE A-1

FORECAST OF SHIPMENTS OF AIR CONDITIONING UNITS FOR 1983

Tons	Projected No. of Units	Projected Total Dollar Value
2	135,000	$ 68,700,000
3	235,000	164,500,000
4	140,000	133,000,000
5	103,000	113,300,000
7-1/2	43,600	78,000,000
10	21,700	50,000,000
15	7,700	27,000,000
20	4,250	17,000,000
25	5,500	33,000,000
50	2,350	25,000,000
75	2,050	32,800,000

and blower section from current Dunham-Bush gas-electric air conditioning units. There would be no necessity for the installer to break the integrity of the refrigerant loop during installation, thus eliminating the possibility of contamination of the system. Installation would require connection of electric supply, solar and cooling tower water, flue pipe, and air distribution ducts. This type of package could be installed in a basement, crawl space, rooftop, or on a slab external to the building.

COMMERCIAL UNIT

The same reasoning applies for the selection of the 10-ton single-package unit as representative of the commercial type of heating and cooling equipment. A 7-1/2-ton size is the lowest tonnage usually included in commercial type

equipment, and more units are sold in this size than any other. The 10-ton size is second in number and dollar volume; however, it is more closely related to the larger sizes in design features. Above 10 tons, both quantity sold and total dollar volume decrease rapidly.

Results obtained from the 10-ton unit test will be applicable to the 7-1/2-ton size and also will demonstrate the feasibility of 15-, 20-, and 25-ton units using direct expansion coils. Only a change in design parameters would be necessary to put the other sizes into production.

MULTIFAMILY AND COMMERCIAL PACKAGED CHILLED OR HOT WATER UNITS

The survey shows that the potential volume for 75-ton units is only 35 percent of the potential volume for 25-ton units. In the original proposal, the 25- and 75-ton units were to have been of the same general design, the basic difference being only in size. In view of this and the fact that the 25-ton unit will adequately demonstrate the feasibility of 50- and 75-ton units, the 25-ton size was selected for prototypes for solar heating and cooling multi-family buildings as well as commercial buildings. The 3- and 10-ton types will air condition spaces by delivering heated or cooled air to the area to be conditioned. The 25-ton unit will provide chilled or hot water to air handlers in the area to be conditioned and provide the individual control necessary for heating or cooling multiple spaces such as with multifamily dwellings.

This heat pump would be installed preferably in an equipment room but could be installed on a slab adjacent to the building with adequate protection from the elements.

GENERAL

Load requirements between and above the selected prototype sizes could be met by using multiples or combinations; for example, two 10-ton units for a

20-ton load, a 10-ton and a 25-ton unit for a 35-ton load, two 25-ton units for a 50-ton load, etc. Using multiple units has advantages such as easier and better distribution without extensive duct work, better matching of load, loss of only part of air conditioning in the event that maintenance is required, and sometimes lower installation costs.

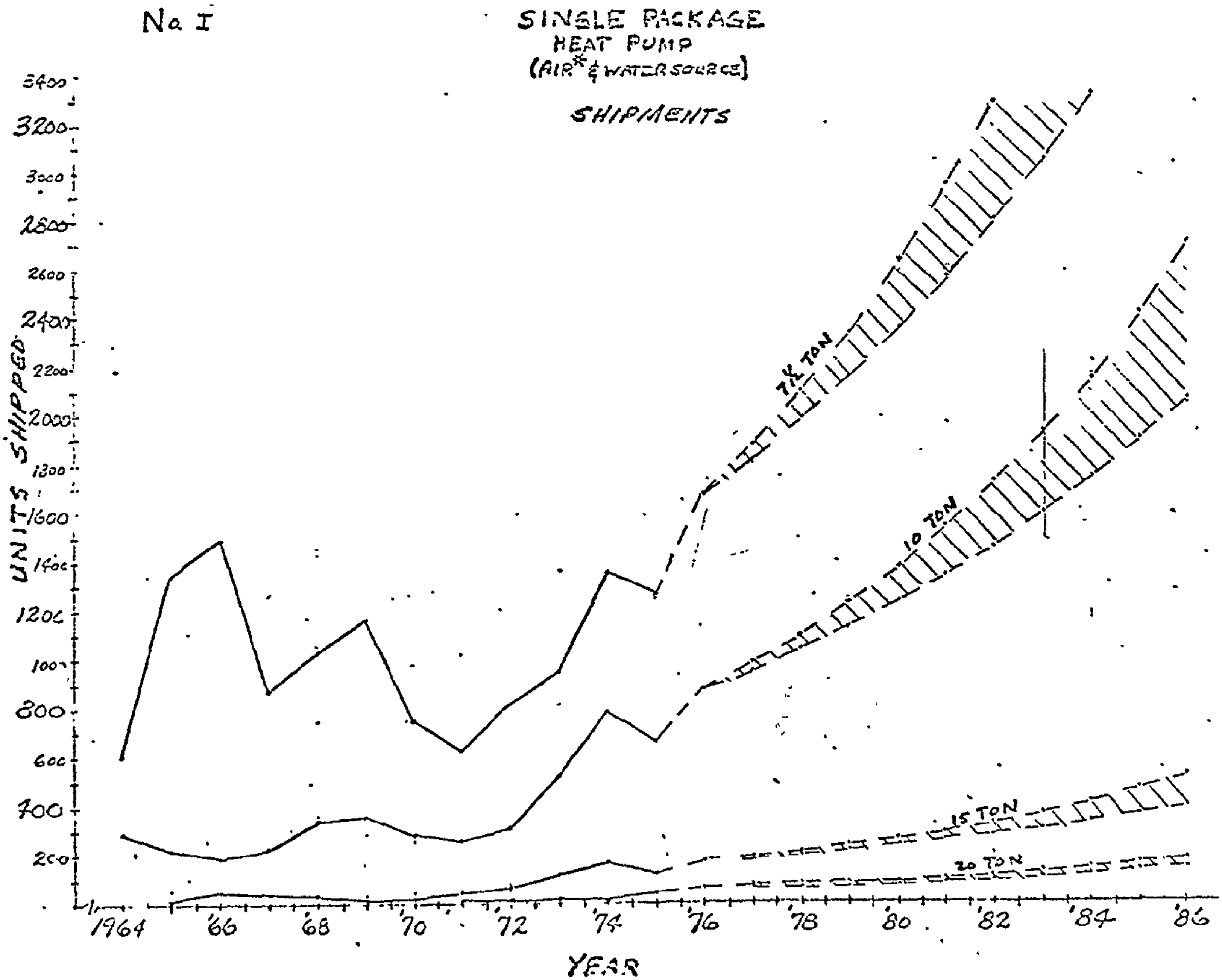

Figure A-1. Single Package Heat Pump (Air and Water Source) Shipments

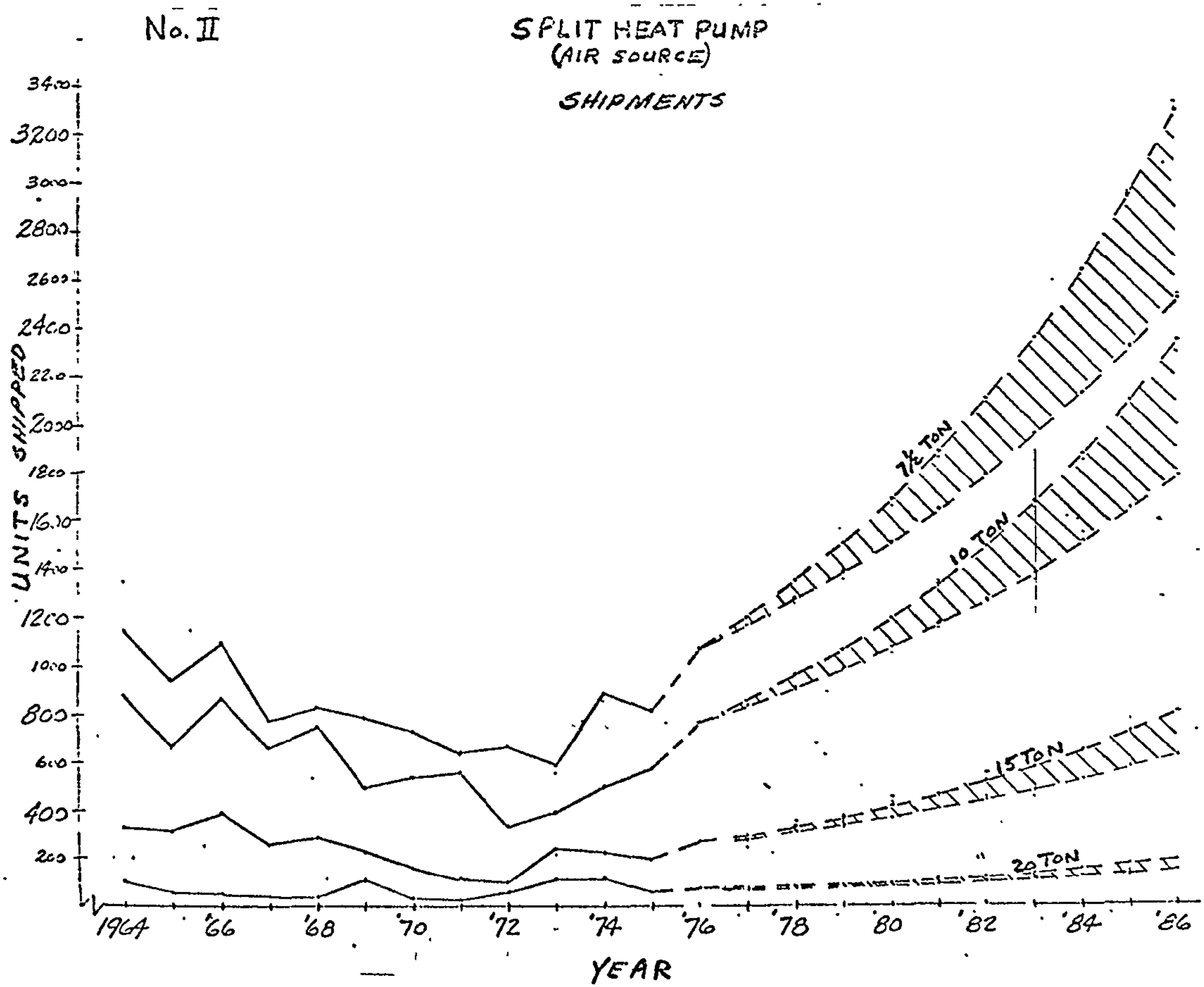

Figure A-2. Split Heat Pump (Air Source) Shipments

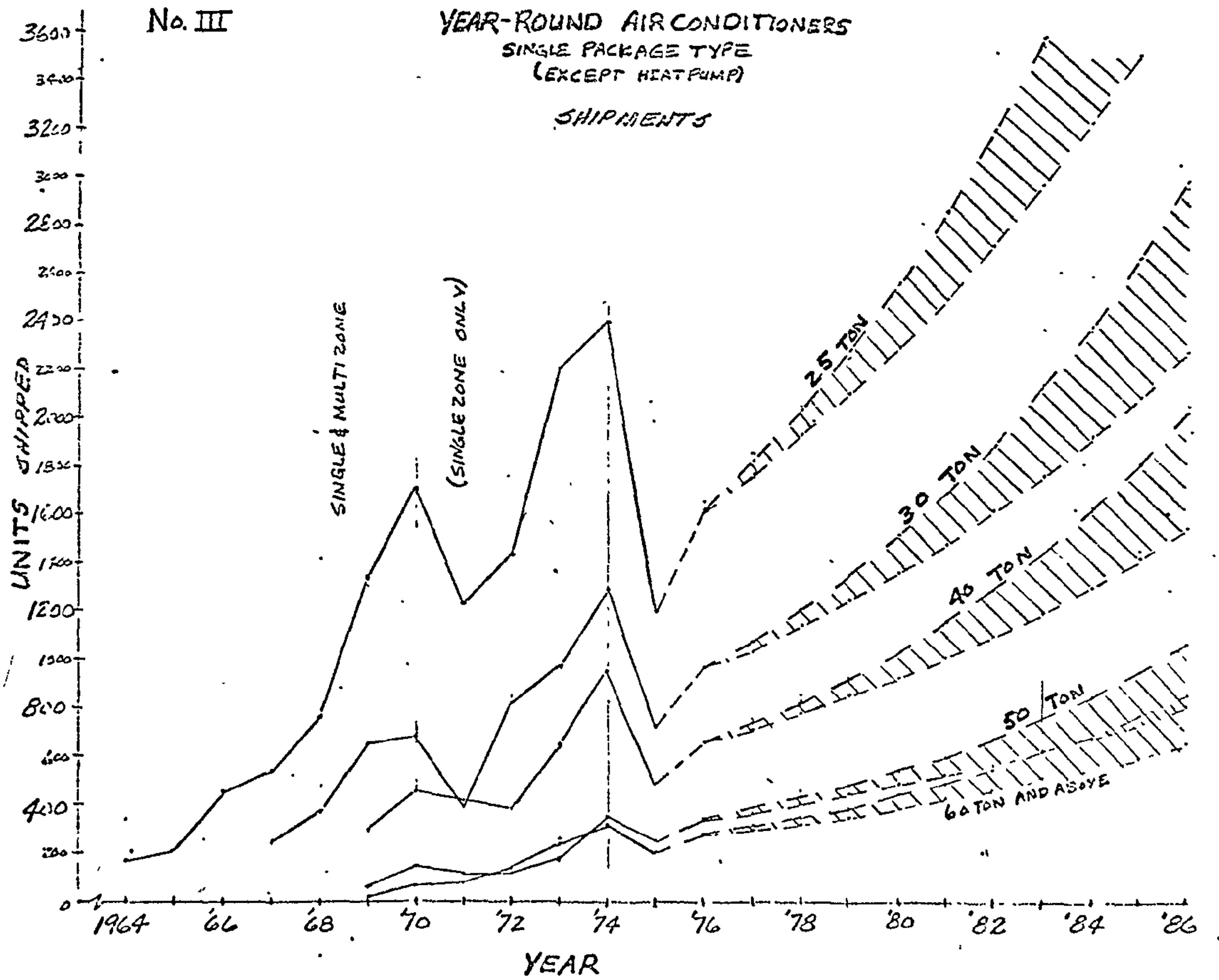

Figure A-3. Year-Round Air Conditioners, Single Package Type (Except Heat Pump) Shipments

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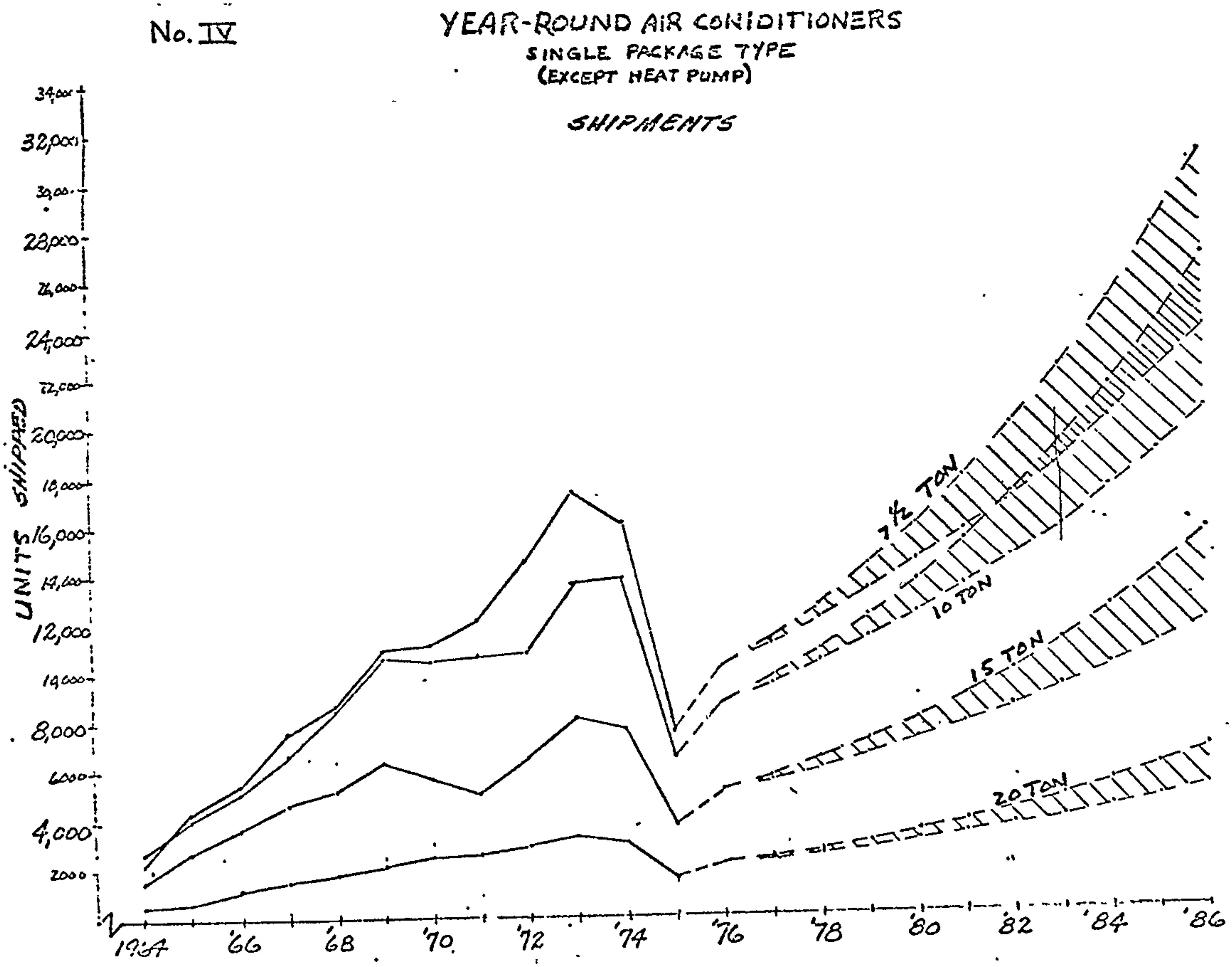

Figure A-4. Year-Round Air Conditioners Single Package Type (Except Heat Pump) Shipments

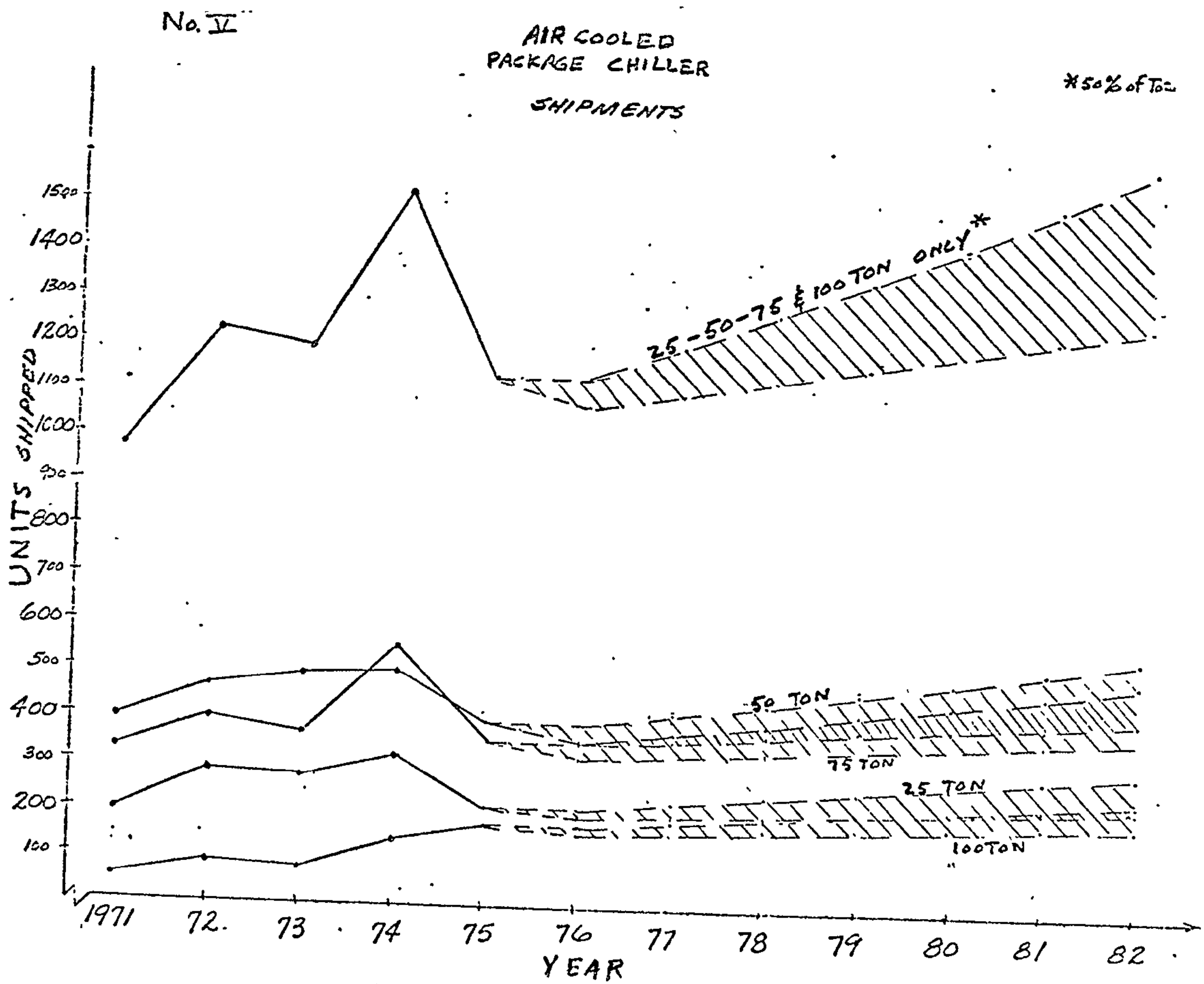

Figure A-5. Air Cooled Package Chiller Shipments

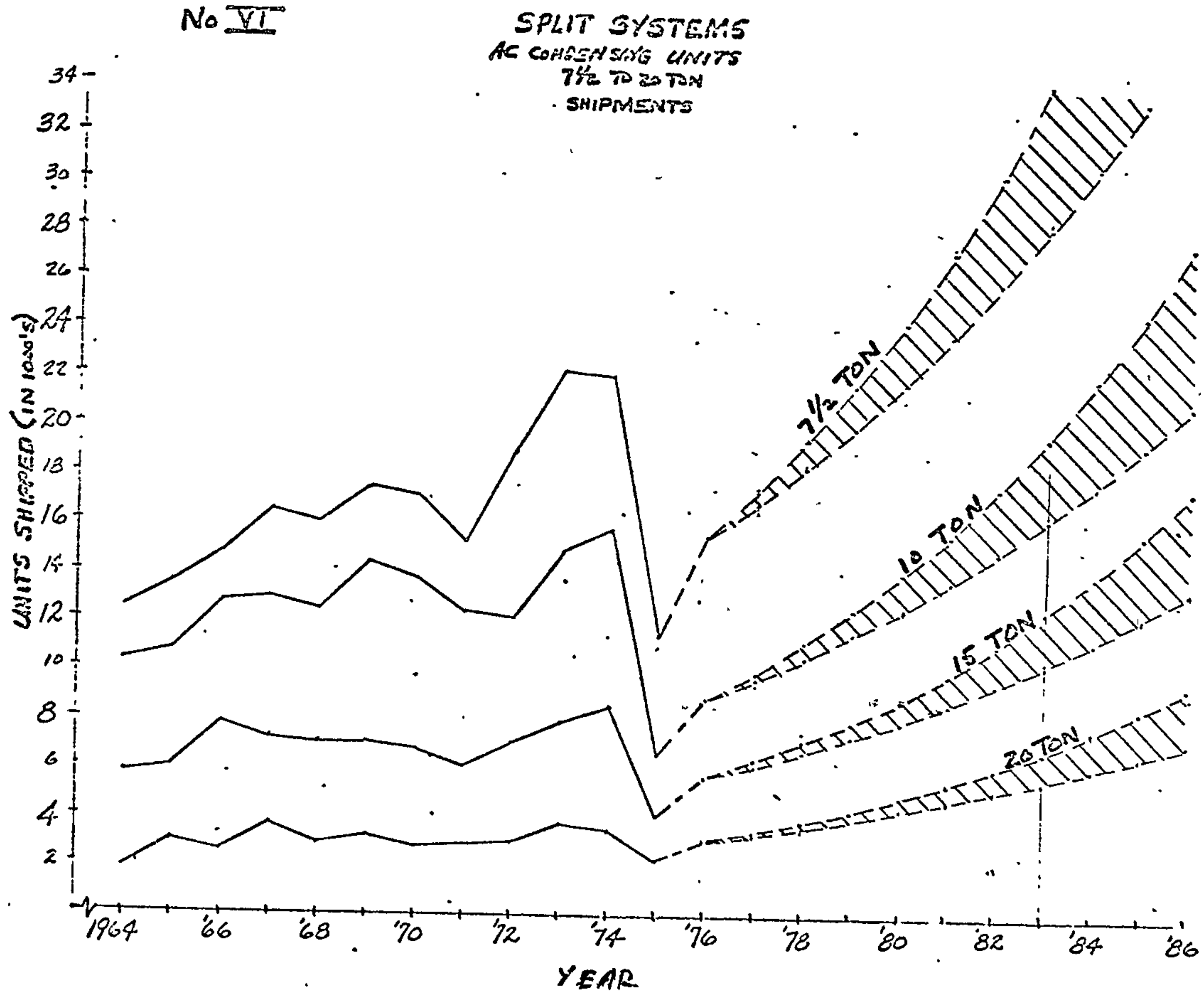

Figure A-6. Split Systems (Ac Condensing Units, 7-1/2 to 20 Ton) Shipments

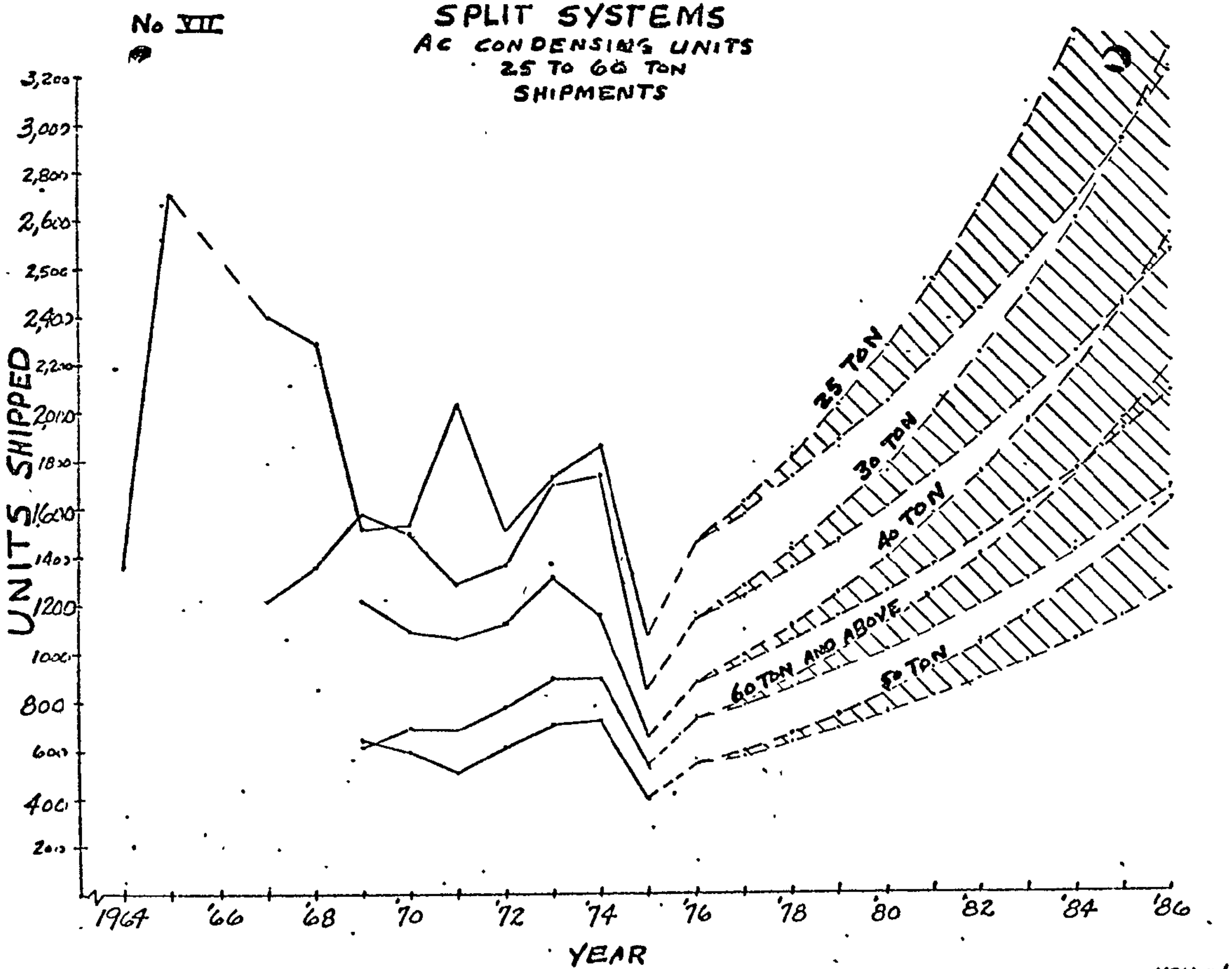

Figure A-7. Split Systems (Ac Condensing Units, 25 to 60 Ton) Shipments

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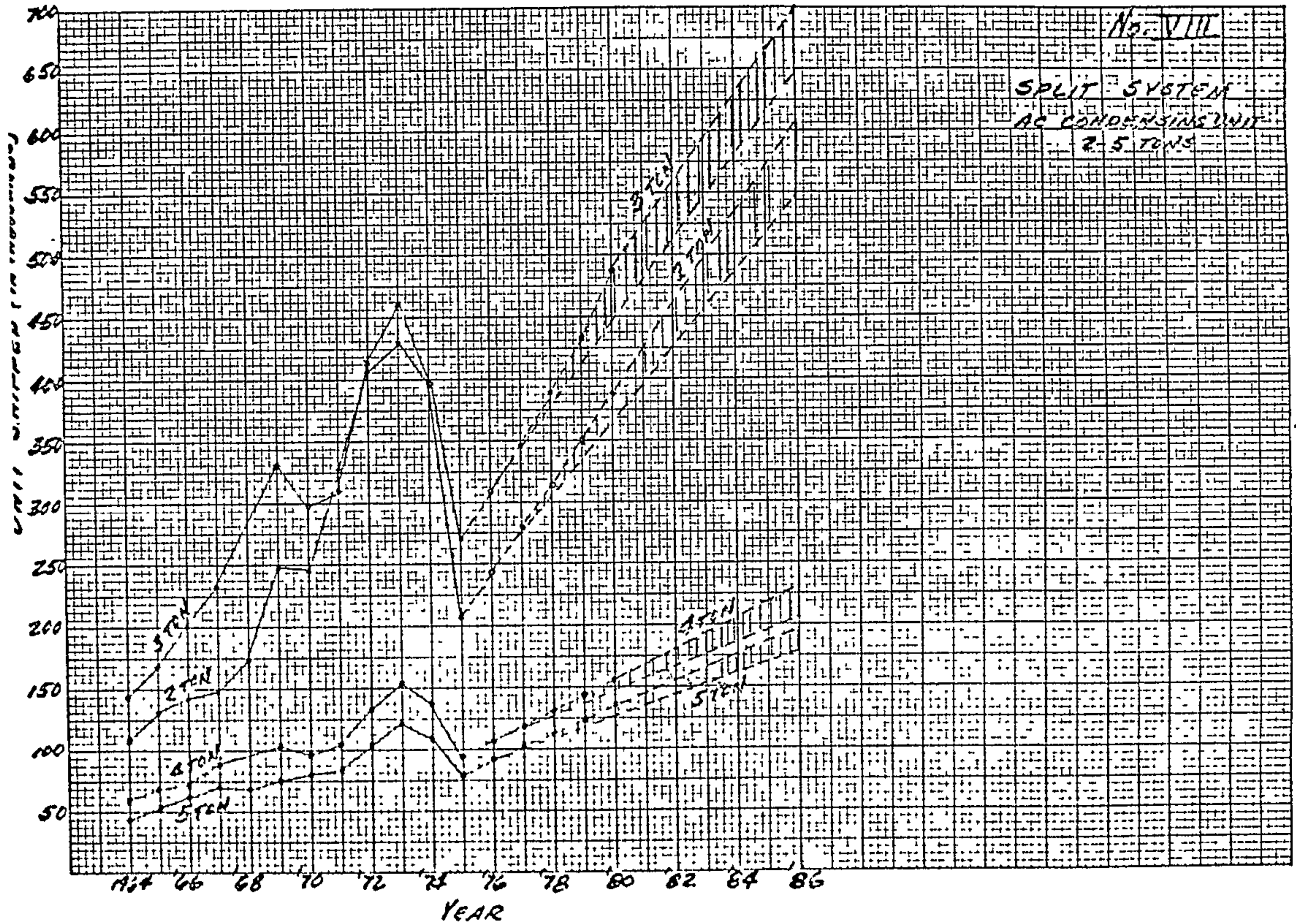

Figure A-8. Split System (Ac Condensing Units, 2 to 5 Ton) Shipments

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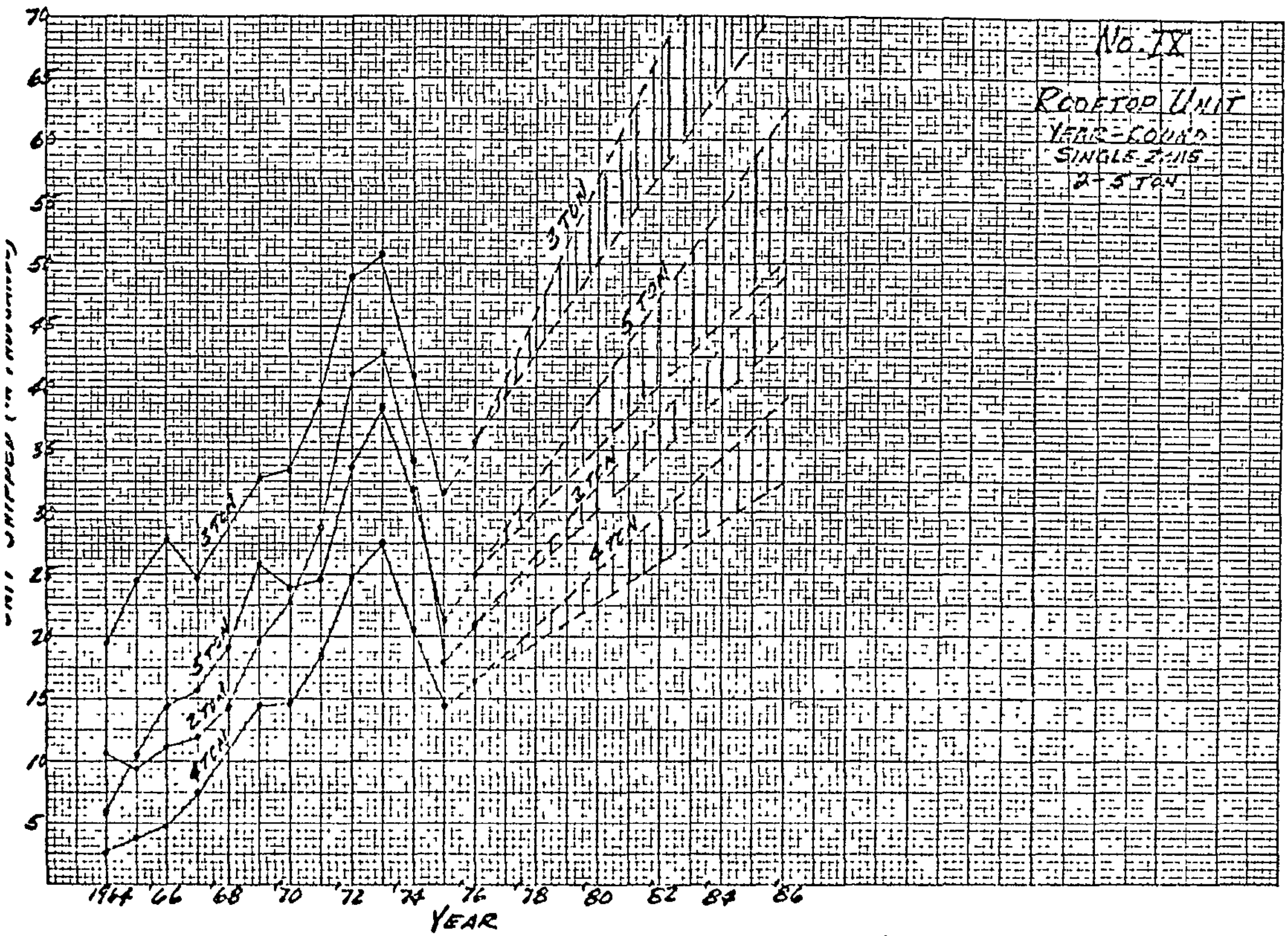

Figure A-9. Rooftop Unit (Year-Round, Single Zone, 2 to 5 Ton) Shipments

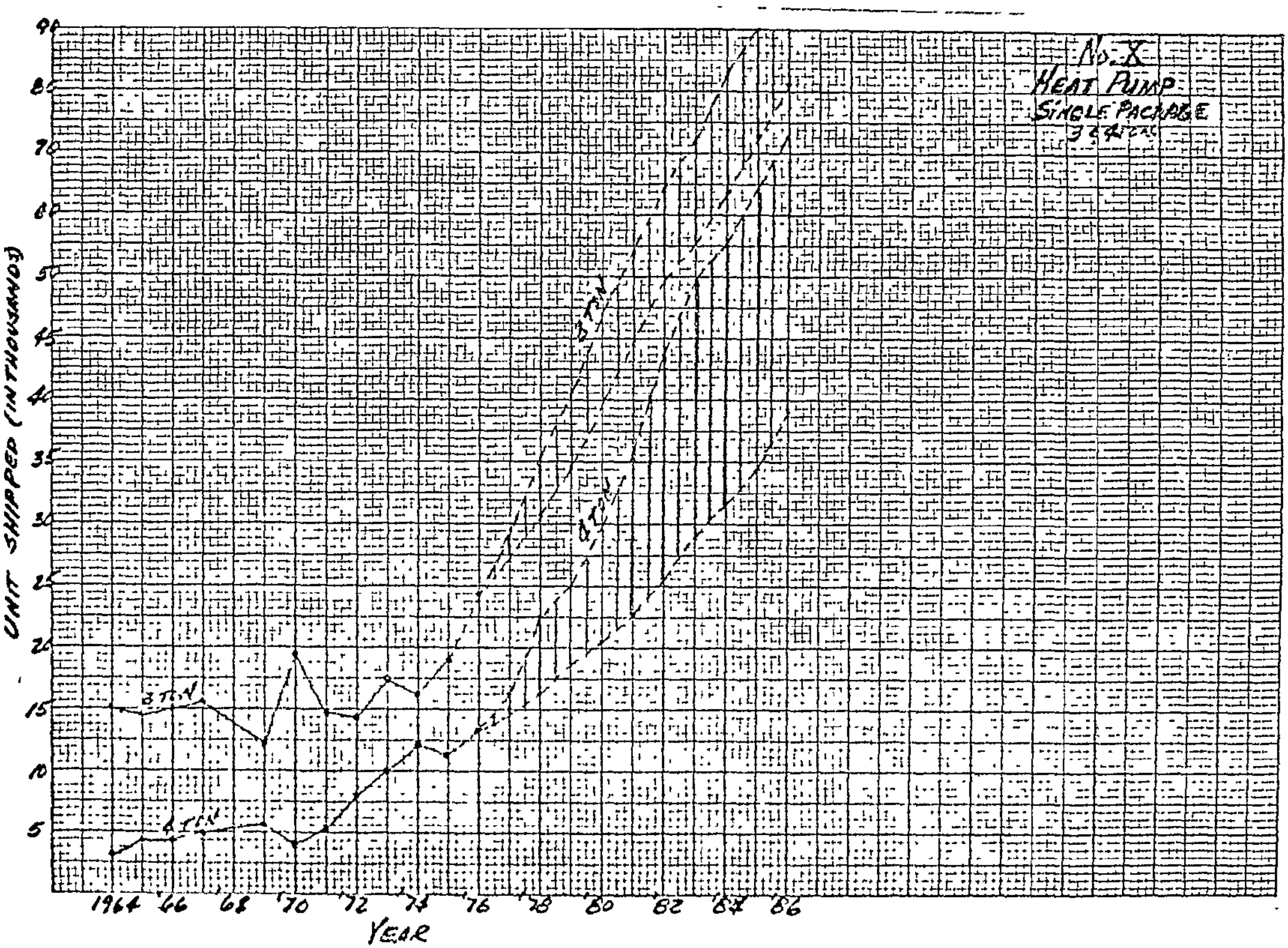

Figure A-10. Heat Pump Single Package
(3 and 4 Tons) Shipments

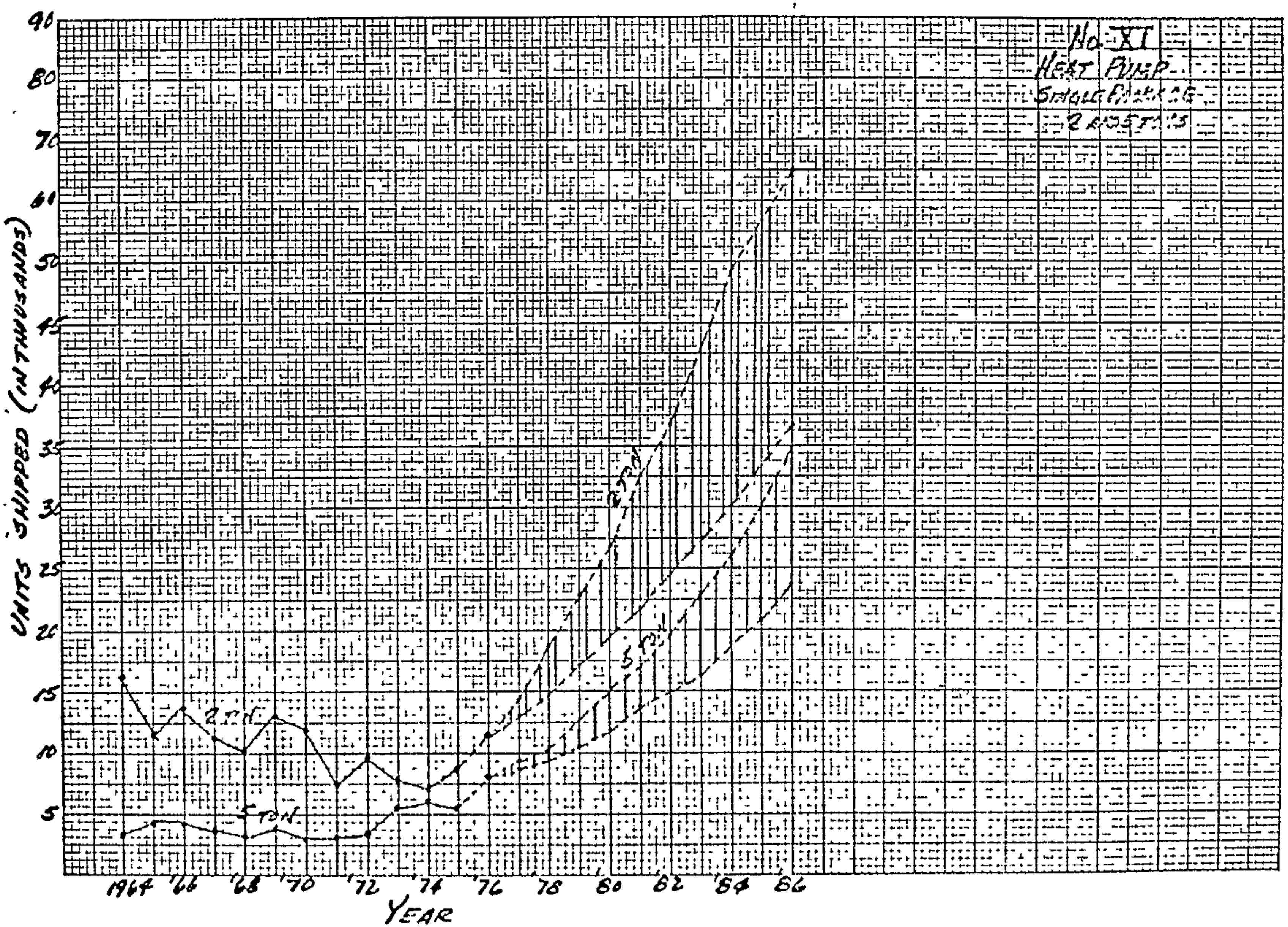

Figure A-11. Heat Pump Single Package
(2 and 5 Tons) Shipments

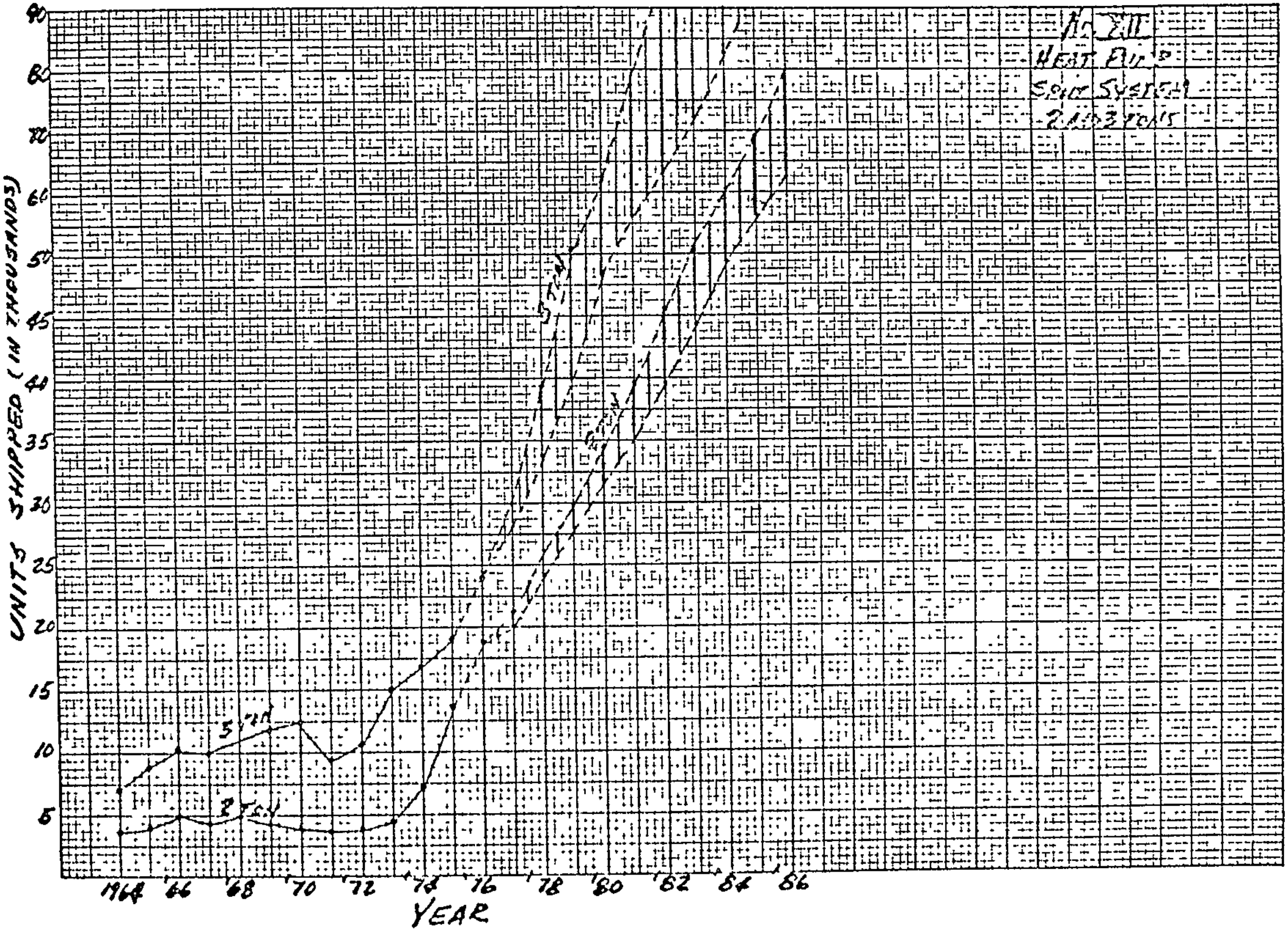

Figure A-12. Heat Pump Split System
(2 and 3 Tons) Shipments

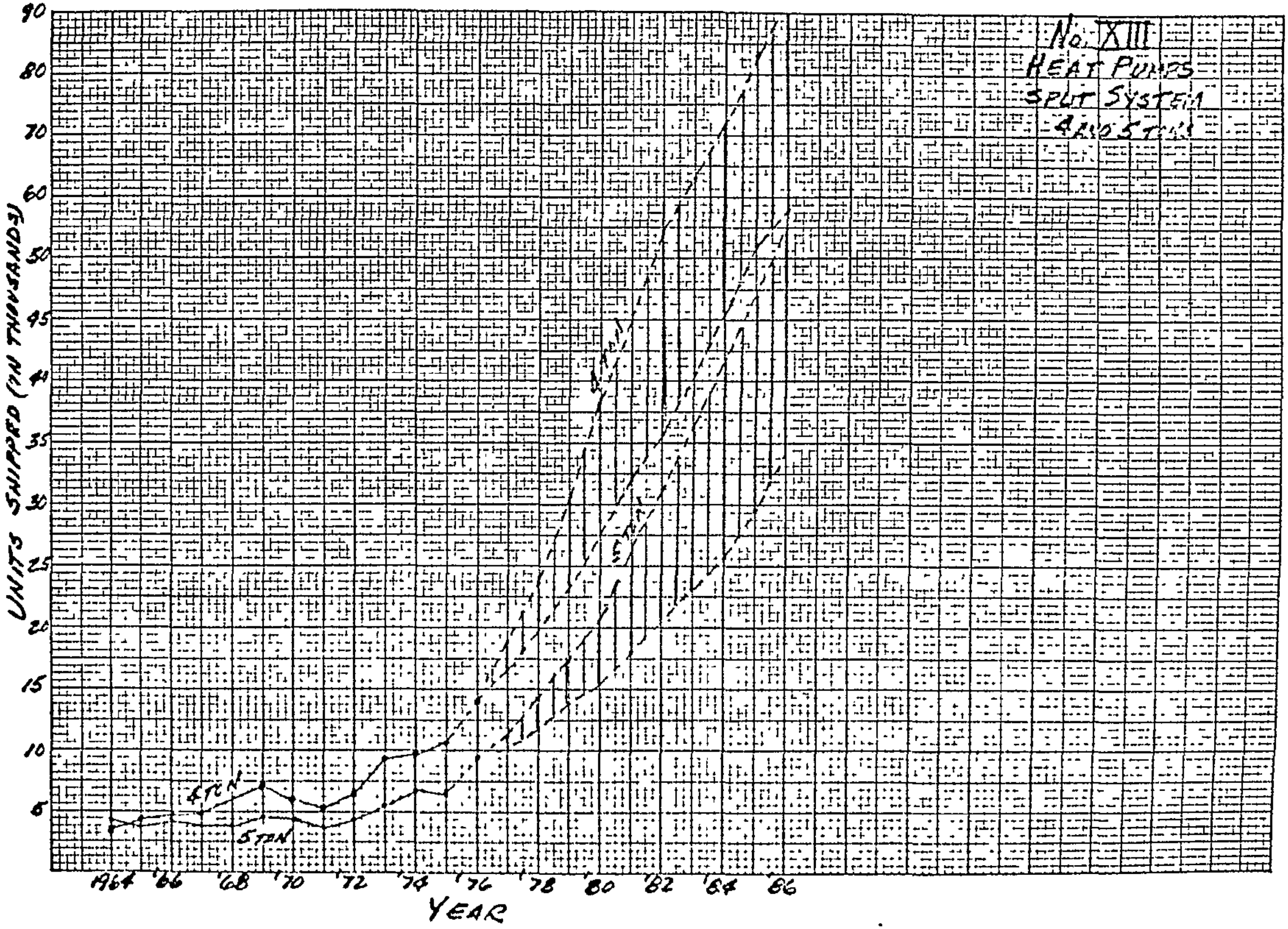

Figure A-13. Heat Pump Split System
(4 and 5 Tons) Shipments

APPENDIX B

HEAT EXCHANGER SIZING

APPENDIX B

HEAT EXCHANGER SIZING

A computer program is used by AiResearch to size plain tube, disk finned tube, continuous plate type finned tube, and wavy continuous plate type finned tube heat exchangers. The tubes may have straight internal fins, an inner fin, or no fins at all. The program considers either straight or curved tubes. The program normally calculates heat transfer between a gas flow over the outside of the tubes with a fluid inside the tubes which can be either single phase, boiling, or condensing. In single-phase heat transfer calculations, the program considers only a single heat exchanger; but with boiling or condensing, the program considers three heat exchanger sections (a vapor region, a phase change region, and a liquid region). The program sizes the heat exchangers for N-pass cross-counterflow of the fluid inside the tubes and straight flow of the fluid outside the tubes. The flow routing may also be folded if required.

Shell and tube heat exchangers with a liquid flowing over the tubes also can be sized by the program. In this case, the fluid inside the tubes can be single phase, boiling, or condensing; and fins may be used on both sides of the tube as described above.

The heat transfer and pressure drop outside the tubes is calculated from Colburn j-factor and Fanning friction factor data input as a function of Reynolds number in a table. This generalized form of input allows any type of fluid to be used outside the tube. Data for typical Dunham-Bush fan coil units are shown in Figures B-1 through B-4. The fin effectiveness is calculated by the program for annular fins of a constant thickness. The

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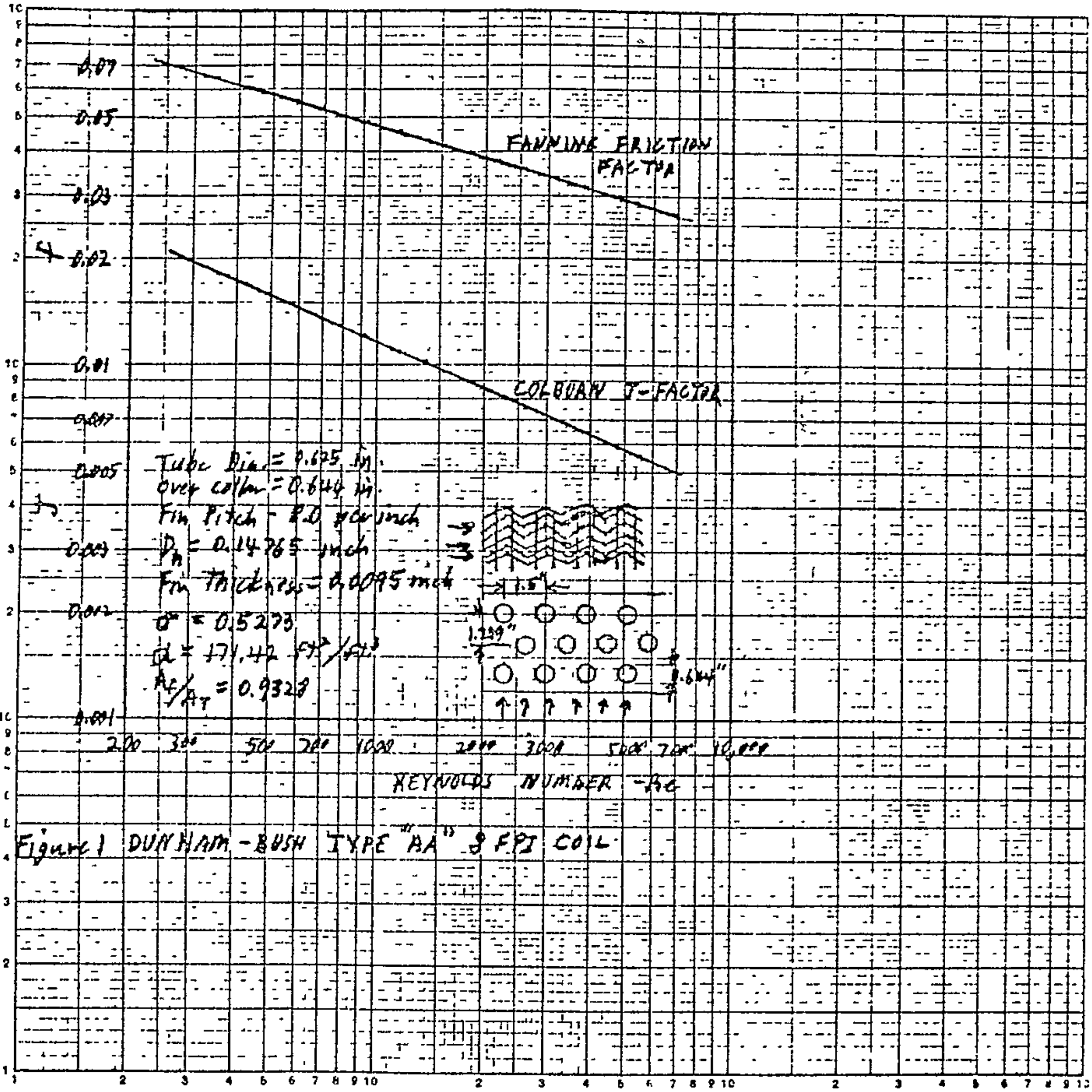

Figure B-1. Dunham-Bush Type AA 8 FPI Coil

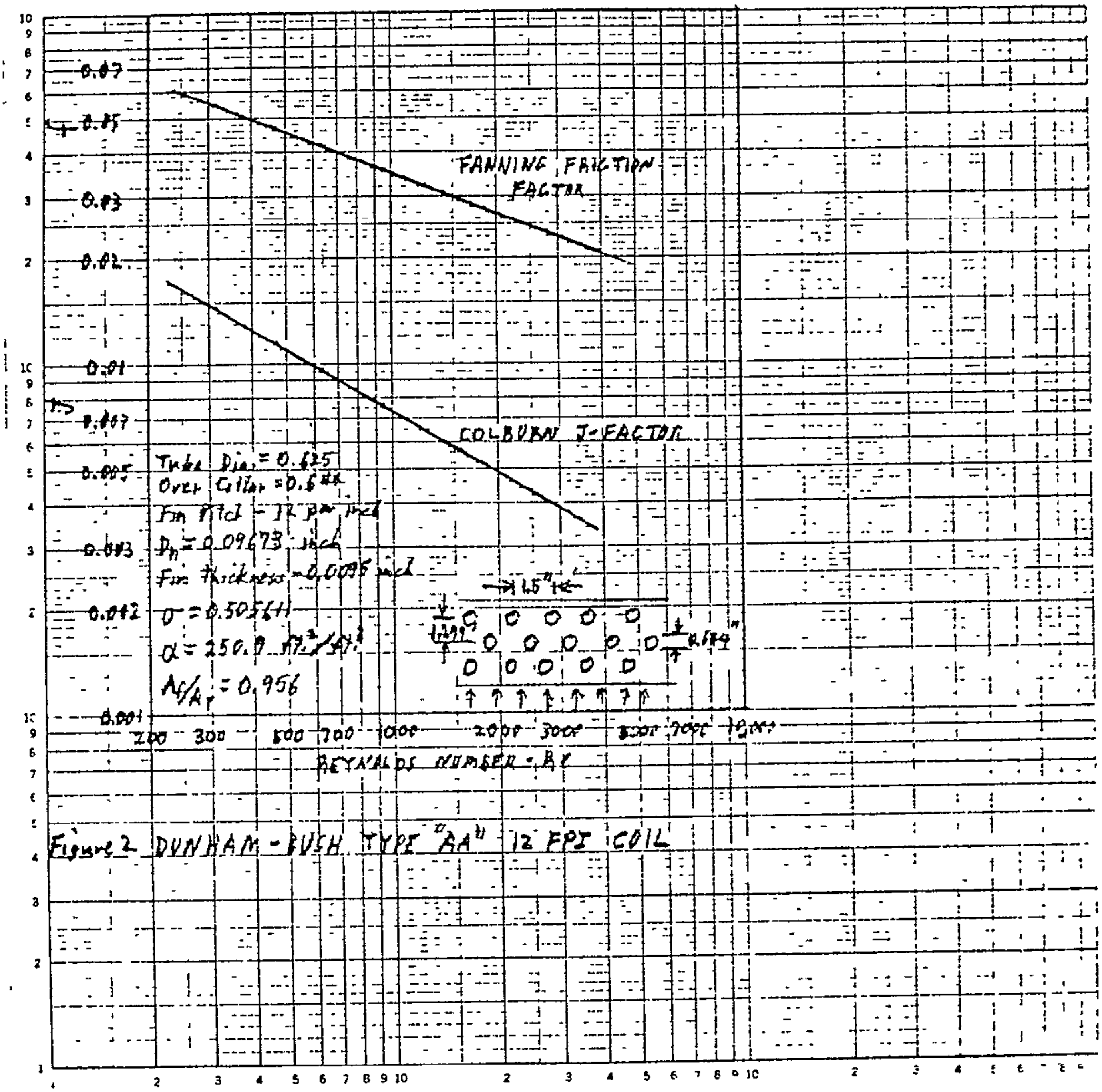

Figure B-2. Dunham-Bush Type AA 12 FPI Coil

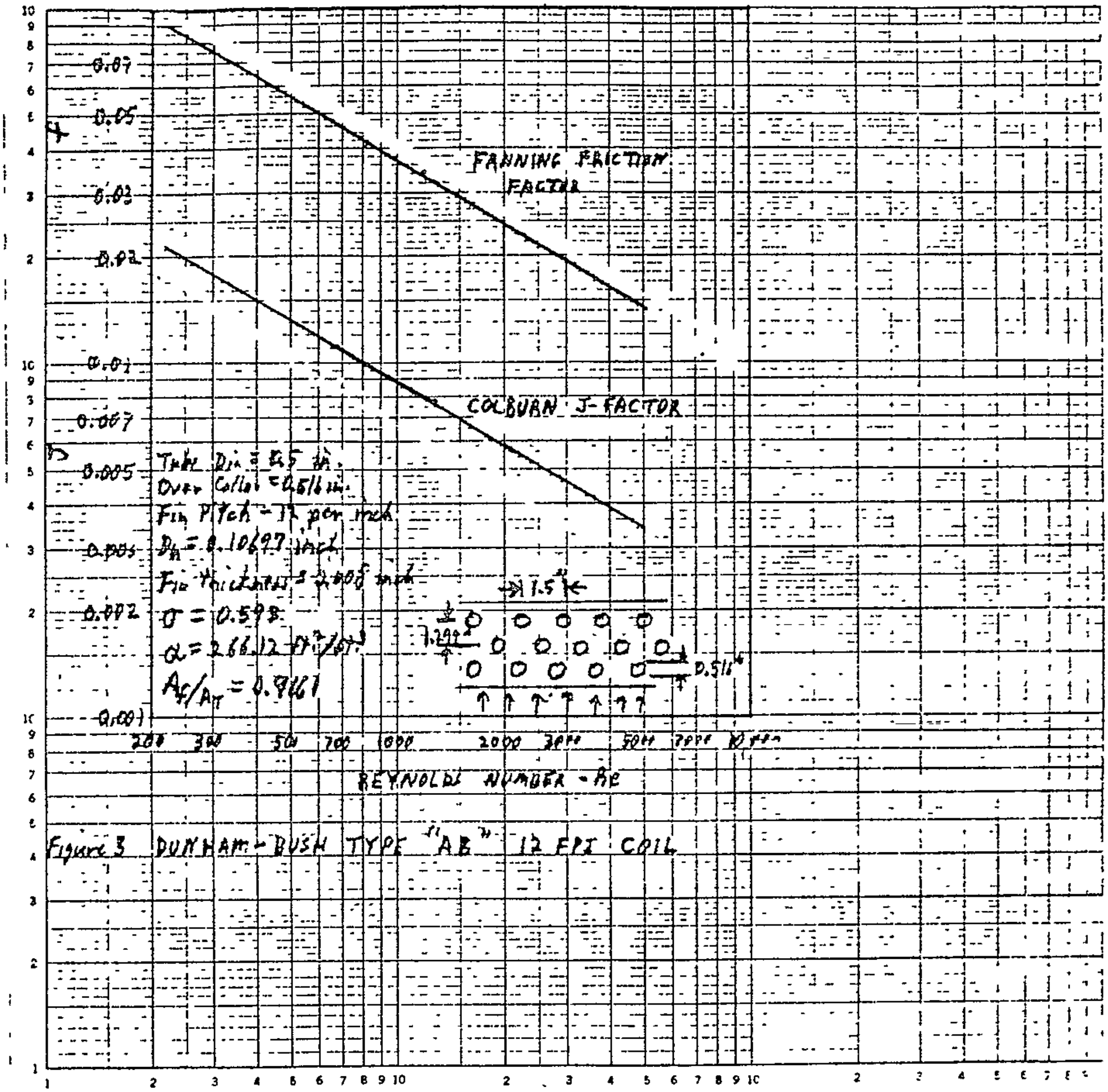

Figure B-3. Dunham-Bush Type AB 12 FPI Coil

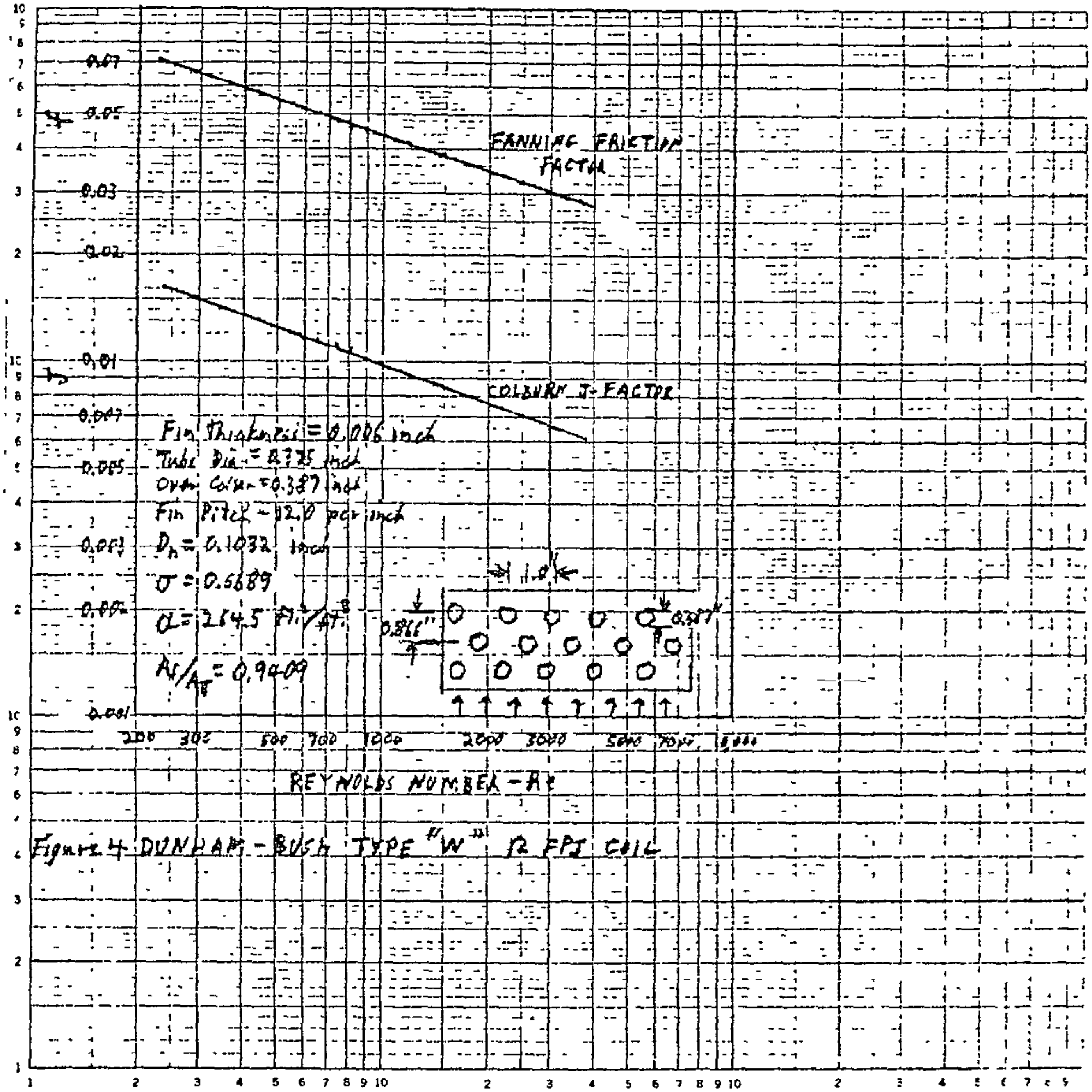

Figure B-4. Dunham-Bush Type W 12 FPI Coil

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heat transfer and pressure drop inside the tubes may be calculated using the normal equations for straight or curved circular tubes, or from input Fanning friction factor and Colburn J-factor data as a function of Reynolds number.

Fluid property data is input in tabular form as a function of temperature, and the program interpolates these tables using a Lagrange interpolation formula.

For boiling heat transfer one of the six following techniques may be selected by the program user.

(a) Boiling by Guerrieri and Talty (Reference 3)

(b) Boiling by Dengler and Addams (Reference 4)

(c) Boiling upflow by Thorsen, Dobran, and Alcorta (Reference 5)

(d) Boiling downflow by Thorsen, Dobran, and Alcorta (Reference 5)

(e) Boiling by John C. Chen (Reference 6)

(f) Boiling by Altman, Norris, and Staub (Reference 7)

For condensation heat transfer one of the two following techniques may be selected by the program user:

(a) Condensation inside vertical tubes (Reference 1)

(b) Condensation inside horizontal tubes (Reference 2)

The fin effectiveness of the internal fins is evaluated for straight fins of constant thickness. The heat exchangers are designed with a margin to compensate for fouling in service and mechanical joint thermal resistance.

The heat exchangers for the 3-ton, 10-ton, and 25-ton heating and cooling systems were sized using this program for the fan coil units, evaporators, and interchangers.

A computer program also is used by AiResearch to size plain tube, finned tube, and corrugated tube condensers. The tubes may have straight internal

fins, an inner fin, or no fins at all. The program calculates the condensing side heat transfer coefficient considering the effects of vapor diffusion to the tube, film type condensation, liquid reheat between tubes, and high vapor velocity over the tubes. The fin effectiveness is included for finned tubes on the condensing side. Either liquids or gasses may flow through the tubes; and fin effectiveness is included for internal fins if they are used.

REFERENCES

1. General Electric, "Heat Transfer Data Book, Vol. 1," Section 506.3, G.E. Research and Development Center, Schenectady, New York.
2. W. W. Akers and H. F. Rosson, "Condensation Inside a Horizontal Tube," Heat Transfer-Stores, Chemical Engineering Progress Symposium Series, No. 30, Vol. 56.
3. S. A. Guerrieri and R. D. Talty, "A Study of Heat Transfer to Organic Liquids in Single-Tube, Natural-Circulation, Vertical-Tube Boiler," Chemical Engineering Progress Symposium Series, No. 18, Vol. 52, 1956.
4. C. E. Dengler and J. N. Addams, "Heat Transfer Mechanism for Vaporization of Water in a Vertical Tube," Chemical Engineering Progress Symposium Series, No. 18, Vol. 52, 1956.
5. R. S. Thorsen, F. Dobran, and J. A. Alcorta, "A Comparative Study of Vertical Upflow and Downflow in a Uniformly Heated Boiling Fluid."
6. J. C. Chen, "A Correlation for Boiling Heat Transfer to Saturated Fluids in Convective Flow," ASME Paper No. 63-HT-34, August 1963.
7. M. Altman, R. H. Norris, and F. W. Staub, "Local and Average Heat Transfer and Pressure Drop for Refrigerants Evaporating in Horizontal Tubes," ASME Journal of Heat Transfer, August 1960, pp. 189-198 (B. Pierre's correlations used and compared with R-22, R-12, and R-11 data).

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